INTERMEDIATE VIBRATION ANALYSIS

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"ONE STEP IN YOUR JOURNEY TO BENCHMARK STATUS"

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Content for this manual provided by CSI Training Instructor(s).
Formatting and editing by M. Angela West.

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TRAINING

Intermediate Vibration Analysis
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Section One
Analytical Troubleshooting

Objectives

- Understand the importance of frequency analysis in determining machinery defects.
- Recognize the importance of predetermining potential defect frequencies.
- Identify the difference between synchronous, nonsynchronous and subsynchronous faults.

Preparing for Analysis

Gathering as much history and physical information as possible before acquiring and analyzing data proves helpful. Knowing the physical specifications of the machine provides the analyst with the information needed to calculate potential fault frequencies. Bearing geometry, coupling type, number of gear teeth, and process considerations all play a part in this process. Take the following steps to properly prepare for analysis:

❶ **Collect machine information.**

Collecting the machine information with a form similar to the one in the appendix provides a great deal of reference information for building a measurement point, as well as a mental picture of the forces and defects that could possibly occur.

❷ **Determine appropriate measurement points.**

Sketch the machine train and define the nomenclature for each of the points. If multiple machines are in question, ensure the point descriptions from one machine to the next are consistent and easily understandable. If monitoring systems are already installed, use them and their point descriptions for ease and faster setup.

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❸ Calculate potential fault frequencies.

Every rotating or moving machine component has the potential of failing. For this reason, the fault frequencies should be predetermined. The manner in which the fault is expected to fail should also be taken into account. If a bearing's inner race is the fault condition being defined, the analyst must consider the higher frequencies first. Also consider other similar faults such as stator slot pass, rotor bar pass, and gears.

❹ Determine alarm criteria.

Setting the alarm criteria for a machine is easier than it sounds. The presence of fault frequencies indicates existing faults. With this in mind, set the alarms without existing faults. If other machines of the same type are accessible, draw comparisons to establish a mean value for energy. Finally, alarms should also consider trends. After setting the initial alarm levels, look at trends with respect to the rate of change between readings.

❺ Set priority of potential faults.

Once the potential faults have been identified, establish the occurrence probability for each fault. Then consider the severity of the fault condition. Faults that may not happen frequently may be considerably more serious, therefore, warranting a higher priority. Last of all, consider the difficulty in detecting the fault condition. If the fault is difficult to identify, the priority for the fault should be placed close to the top of the list.

❻ Determine possible fault causes.

Most faults have a variety of possible causes. For example, unbalance may be caused by material build up, wear, broken components, etc. After each of the different causes have been identified, prioritize each cause for each fault condition.

❼ Establish preventive actions.

Some preventive actions should be established to eliminate or at least minimize the frequency of occurrence. Balance and alignment problems can be minimized by precision balancing and alignment techniques.

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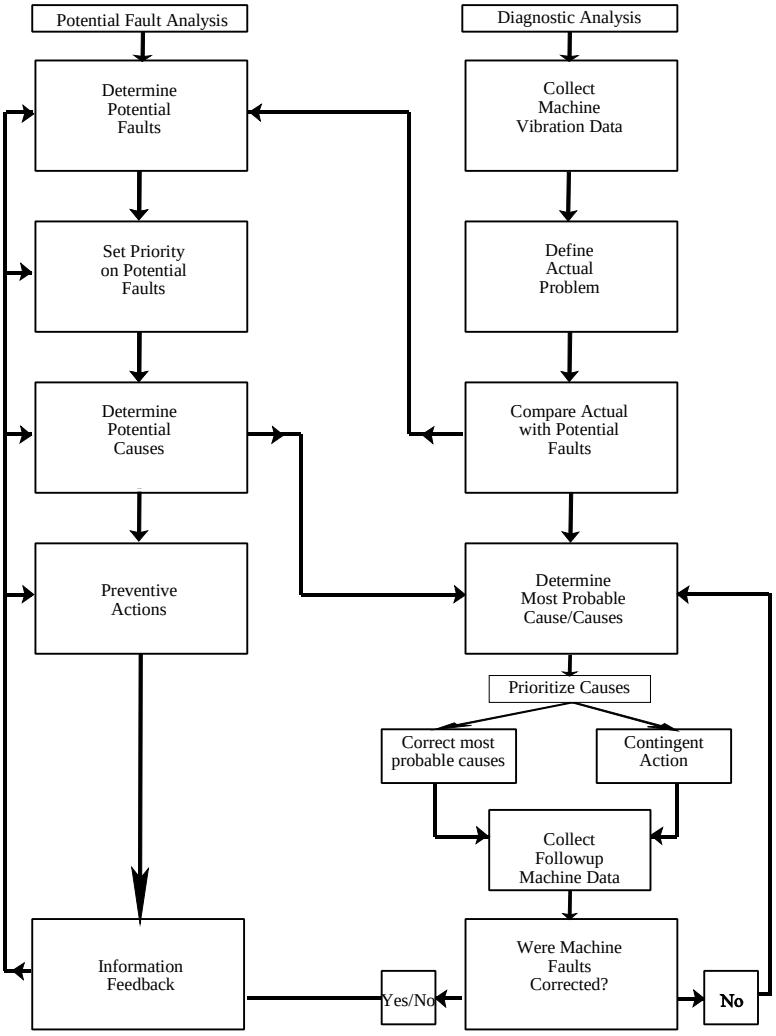
❽ Establish information feedback.

After all is said and done, a feedback loop must be established to refine all the information above. Spectral and waveform data, trends, other machine changes, operational speed and load, work and the reasons for the work should all be a part of the feedback loop. The chart on the next page illustrates the feedback loop. Other components and potential faults should also be considered and prioritized.

Vibration Analysis Flow Chart



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Steps to Solving Vibration Problems

❶ To identify the problem causing the machine vibration, ask yourself some questions.

- What part of the machine has the vibration problem?
- How was the vibration measured?
- Were good measurement procedures used?
- Is the data valid?
- Does the vibration problem occur at only certain loads, temperature or power conditions?
- Is the machine also noisy?

❷ The machine geometry should be understood as completely as possible.

- Sketch the entire machine train.
- Identify all the major components: motor, pump, gears, etc.
- Identify specifications on all the bearings in the machine.
  - Sleeve or antifriction
  - Type
  - Number
  - Bearing geometry
- Determine belt information.
  - Center-to-center distance
  - Pulley pitch diameters
  - Number of belts
- Couplings.
  - Gears
    - Spool length
    - Lube practice
    - Clearances
    - How aligned

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- Disc

  - How aligned
  - End clearances
  - Maintenance practices

- Bun

  - How aligned
  - Bun compound
  - Possible torsional problems

- Fluid

  - How aligned
  - Estimate slippage

- Drives: motors, engines, turbines
- Gears: types and reduction ratios
- Shaft diameters and lengths
- Rotor dimensions and weights
- Other information unique to the machine

❸ Gather available maintenance history and any previous vibration data.

- Has vibration data been collected on this machine before?
- Has the vibration fault been developing over time, or is it a new problem?
- Determine the last thing done to the machine.
- Talk to the machine operator and get his/her input.

❹ Determine the best points to collect data.

- At the bearings
- In the problem area
- Consider possible resonances
- Consider forcing functions from other machines
- What type of instrumentation will the solution of this vibration problem require?
  - Tape recorder
  - Impact data
  - Noncontact probes
  - Displacement, velocity, or acceleration probes

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- Coastdown or startup data
- Single or multichannel data
- Reference transducer input
- Current transformer
- Special averaging methods
- Temperature data

❺ Determine as many as possible forcing frequencies before taking data.

- Determine any and all shaft rotational speeds. Most machine defect frequencies are related to a shaft turning speed.
- Bearing fault frequencies - BPFO, BPFI, FTF, and BSF.
- Belt frequencies.
- Gear Mesh frequencies.
- Blade pass frequencies.
- Resonant frequencies.

❻ Take data.

- Frequency data, at least horizontal, vertical, and axial data at each bearing. It may be helpful to compare normally averaged data with synchronous time averaging to identify synchronous and nonsynchronous components.

- Check for beats by watching the instantaneous spectrum or by comparing the peak hold spectrum with the normally averaged spectrum.

- Check the skirt width of the spectral components. Steady state signals collected with a Hanning window will occupy three to four cells. Wide skirt widths indicate the presence of signal modulation, another component very close in frequency, or a component that is varying in frequency during the sample time.

- Do not limit your data collection to the bearing locations. Data at the bearings should be considered the minimum data to be collected. Consider data collection on the machine case, foundation, piping, etc.

- Once data has been collected, break the spectrum into three different regions: subsynchronous, synchronous, and nonsynchronous.

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FAULT GUIDE

Vibration	Dominant Frequency	Dominant Plane	Phase Reading
Unbalance			
Static	1xTS	Radial	Radial in phase
Dynamic	1xTS	Radial	Radial 0-180 out / 2 plane
Couple	1xTS	Radial/axial	Radial 180 out
Overhung rotor	1xTS	Radial/axial	Radial unsteady/Axial in phase
Misalignment			
Angular	1x, 2xTS	Axial	Axial 180 out
Offset	1x, 2x, 3xTS	Radial	Radial 180 out
Offset + angular	1x, 2xTS	Radial/axial	Radial/axial 180 out
Sleeve bearing	1x, 2xTS	Radial/axial	Axial 180 out
Antifriction bearing	1x, 2x, 3xTS # of balls x TS	Axial	Axial 180 out
Bent Shaft	1x, 2xTS if on coupling end	Axial	Axial 180 out
Mechanical Looseness			
Non-rotating bearings	1 - 10xTS	Radial	
Rotating impellers	1xTS predominant as high as 10xTS		
Antifriction Bearings - (early stages)	Bearing frequency	Radial	
Antifriction Bearings - (late stages)	1xTS and harmonics	Axial on thrust bearing	
Sleeve Bearings			
Looseness	Multiples of TS	Radial	
Oil whirl	0.43 x TS	Radial	
Belt Drives			
Mismatched, worn	2 x belt frequency	Radial inline with belt	
Eccentric sheave	1 x shaft speed	Radial	
Misalignment	1xTS	Axial	
Gears — GMF = Gear Mesh Frequency, SG = Spur Gears, HE = Helical Gears			
Transmission error	GMF 1 + harmonics	Radial SG / axial HE	
Pitch line runout	GMF + sidebands	Radial SG / axial HE	
Unbalance	1xTS	Radial SG / axial HE	
Misalignment	1x, 2xTS	Radial SG / axial HE	
Faulty tooth	GMF + sidebands	Radial SG / axial HE	
Rotor Rub	0.5xTS and ½ multiples	Radial	
Electrical			
Loose iron	2 x line frequency (LF)	Radial	
Stator problems	2 x LF	Radial	
Phase unbalance	2 x LF	Radial	
Loose stator	2 x LF	Radial	
Broken rotor bar	2 x LF at 1xTS with with sidebands	Radial	
Eccentric rotor	2 x LF at 1xTS with slip sidebands	Radial	
Loose slot	2 x LF, slot frequency + sidebands	Radial	
Pole pass	At 1xTS with sideband spacing = to # of poles x slip frequency		
Blade/Vane Pass	# of blades/vanes x TS	Radial	

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Subsynchronous Frequencies

Subsynchronous frequencies appear below the shaft turning frequency.

Oil Whirl

Oil whirl occurs at approximately 44 percent of shaft frequency in pressure fed sleeve bearings. The frequency decreases as the shaft speed slows down. The frequency usually drops out at about 75 percent of normal running frequency.

Oil Whip

Oil whip occurs when the shaft is running at a turning frequency above the second critical frequency. The oil whirl frequency locks onto the first shaft critical frequency turning into oil whip. The frequency does not drop away until the shaft frequency drops below the second critical frequency.

Rub

Rub frequency is close to 50 percent of shaft frequency and 1½, 2½, etc., and harmonics.

Antifriction Bearing Loose in Housing

This frequency is 50 percent of shaft frequency, but 1½, 2½, etc., not as noticeable as rub.

Cage or Train Frequency of Antifriction Bearing

This frequency usually indicates advanced stage of bearing failure. Check for outer race fault and its harmonics as well as sidebands at the cage frequency.

Primary Belt Frequency

Check by calculation and look for higher harmonics. Could be caused by belt misalignment, worn sheaves, or defective belts.

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Defective Tooth-to-tooth Repeat Frequency

Usually a very low frequency which can be better seen in the time domain. Often referred to as a hunting tooth frequency.

Surge

Usually a high component from 10 percent to 50 percent of rotor frequency. Check differential pressure across the fan or pump. Check the operating point with the best efficiency point on the fan or pump curve.

Ignition or Fuel Problem on Four-Cycle Recip

Usually accompanied with higher ½ orders.

Synchronous Frequencies

Frequencies synchronous with the shaft in the bearing where the data is taken. The frequencies are integer multiples of the reference pulse.

Unbalance

One times shaft speed. If any looseness exists in the system, there may be several harmonics of the first order component.

Check horizontal and vertical amplitude and phase at each bearing. If horizontal-to- vertical phase shift is approximately 90° on both sides and the phase relationship is similar, then it is unbalance.

If the horizontal-to-vertical phase shift is not close to 90°, consider possible pedestal resonance or shaft centerline misalignment.

If the phase relationships side-to-side in both the horizontal and vertical directions are not similar, consider the possibility of misalignment.

If the readings indicate an unusually high unbalance, look for a possible resonance, bent shaft, or fault in rotor supports. Use the weight of the rotor in ounces times the vibration amplitude in inches to estimate the unbalance in ounce inches. For example, a 20,000 ounce rotor times an amplitude of 0.005 inches (5 mils) = 100 ounce inches of unbalance.

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If the problem is unbalance and impact data can be taken, impact both sides and estimate the pivot point. If this is far outside the bearings, it may not be possible to field balance the rotor. In addition, the impact data will indicate the first critical at each bearing. If this is close to the running frequency, it may be better to consider stiffening the system.

Misalignment

The first effect of shaft misalignment is an increase in the radial load on the bearings. In most cases, the next effect is an increase in the first order of the shaft frequency. As the condition worsens, the second order builds.

Check the axial vibration on the bearings on each side of the coupling. If in phase, consider balance or gear coupling lock-up.

If the misalignment is primarily angular, the top-to-bottom and the side-to-side phases on each bearing are normally out of phase.

Bent Shaft

High first order, if balance is attempted a bent shaft will usually require large correction weights. Check the axial data at both bearings; 180° phase shift is an indication of a bent shaft. If possible, check the shaft with proximity probes or dial indicators (be sure to check low-speed runout).

Check the top-to-bottom and the side-to-side phase on each bearing. If they are out of phase, it is probably a bent shaft. In most cases of a bent shaft, the end-to-end phase readings of either the horizontal or the vertical radial vibration are the same.

Looseness

Look for many harmonics of shaft frequency. Usually the second is almost as high as or higher than the first order. However, the harmonics will be predominantly odd order if it is pure looseness.

Any truncated function will produce harmonics in the spectra so that a condition where the time domain waveform is nonsymmetrical will look like looseness. Check the time domain.

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One or more system resonances may be excited by one or more of these harmonics so that the levels of the harmonics is magnified by the resonant amplification.

Pitch Line Runout on Belt Sheaves

Often confused with unbalance. Check vibration in line with the belt drive. The frequency of the sheave with the runout will usually appear at the other sheave.

If the belts are removed and the first order vibration is significantly lower, it is not unbalance, but more likely pitch line runout.

The vibration will be at the frequency of the sheave with the pitch line runout.

Cavitation

Look for the first order and higher harmonics up to the number of blades along with very high frequencies. Check the time domain.

Cavitation is the implosion of a void or bubble in the intake fluid when it reaches the pressure side of the pump or fan. Because this can be a very steep waveform, it can usually be easily identified in the time domain. It results in many higher harmonics in the frequency domain. Often the blade pass frequency will appear.

Blade Pass Frequency

Blade Pass Frequency is defined as the number of blades, etc., times the rotational frequency. Blade pass is always there; however, high levels can indicate system resonances excited by the blade pass component or an arrangement of supports which causes process flow variations in sync with the blade pass frequency.

On axial flow units, the primary blade pass may be reduced by an out-of-pitch or out- of-track blade, but other harmonics will be higher. Check bearings in the axial direction for components from the first order up to the number of blades as a possible indication of this condition.

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Gear Mesh Frequency

Gear Mesh Frequency is calculated as the number of gear teeth times the frequency of the gear. These can be very high frequency and many times must be measured with an accelerometer. Frequencies as high as 8,000 to 10,000 Hz are not unusual.

A magnet base cannot be used to collect this data. Because the levels can be as high as 50 to 150 g's, it may be difficult to use a hand-held probe. The transducer should be stud mounted or glued to the gear box if the levels are above 20 g's. In addition to the gear mesh frequency and its harmonics, the pinion shaft, gear shaft, gear wobble, hobbing ghost, tooth resonance, and entrapped oil frequencies and their harmonics may also be present in the spectra.

Nonsynchronous Frequencies

These frequencies are higher than the shaft turning speed, but they are not integer multiples of the shaft turning frequency.

Another Component in the Machine

Identify and determine severity. If possible, attempt to simplify the system by disconnecting or shutting down some part of the whole.

One or More System Resonances

When a system is excited by some energy in the machine or in the process; take enough impact data to identify such resonances and their excitation.

Belts

Even though the primary belt frequencies are subsynchronous, multiples of this frequency can be nonsynchronous. The largest components of belt vibration are usually at higher frequencies. In most cases, the highest level is at the primary belt frequency times the number of pulleys over which the belt passes.

In multiple belt drives, there may be frequencies from two times the primary belt frequency up to twice the number of belts times the primary belt frequency in the spectrum.

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Any of these frequencies may excite a system resonance and cause the vibration level to exceed the acceptable level.

Belt alignment, tension, and quality all play a part in the level of vibration from the belt drive.

In some cases, it is impossible to reduce the belt vibration to an acceptable level. Replacing the regular V-belts with Grip-Twist or Grip-Link V-Belts usually reduces the vibration level from two to four times.

For high quality drives, it may be necessary to use flat belts, such as the Habasit belt, in order to reduce the vibration to an acceptable level. These belts usually reduce the belt vibration ten times or more.

Antifriction Bearings

The basic frequencies generated by antifriction bearings are the cage or carrier, the ball spin, the outer race fault, and the inner race fault. One or more of the primary frequencies along their harmonics and their sidebands may appear in the spectra. Because the outer race is always in the load zone, however, the outer race frequency and its harmonics are the most commonly detected.

The bearing fault frequencies can be calculated accurately from the bearing physical data or estimated. Although the major causes of antifriction bearing failure are improper handling, installation, and lubrication, vibration can give a reliable and early indication of bearing failure.

For antifriction bearing analysis, data should be taken as acceleration using the shortest solid path to the load zone of the bearing for the transducer mounting. Carefully examine the spectra for a series of harmonically related peaks which are not synchronous with the shaft in the bearing.

If the actual fault frequency values are not known, check to see if this series of peaks falls near the approximate value. Operating the bearing above the rated load and at higher than recommended shaft speeds will shorten the life of the bearing under the best of conditions.

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Electrical

Vibration can be caused by the dress of the conductors in the raceway, loose lamination in a transformer or a motor, broken or cracked rotor bars, open shoring rings, eccentric rotors, eccentric stator, etc. Conductors should be redressed and loose lamination should be readjusted and tightened to reduce the line frequency and the two times line frequency components.

Eccentric stators cause two times line frequency vibration, which is directional to the largest air gap.

Magnetic misalignment results in two times line frequency plus the number of poles times the slip frequency sidebands.

Eccentric rotors cause running speed and twice running speed components with the side bands at the number of poles times the slip frequency. Broken or cracked rotor bars or open or shorted rings may cause line frequency and twice line frequency components at the number of poles times the slip frequency. Broken or cracked rotor bars often show up at one times turning speed with sidebands at the number of poles times the slip frequency.

This defect only shows up under load, so many motor repair shops are unable to determine these faults. One or more of the faulty bars will cause unequal heating in the rotor, which results in a thermal vector so that the balance will only be good for one load condition. Such thermal vectors are very common in motors and generators.

Noise

Chain drives, gears, the process, etc., can be the cause of high noise levels in the system. It is important to separate the noise problem from the vibration problems.

In general, noise levels are reduced by sound absorbing materials and vibration levels are reduced by correction.

Often high noise levels do not indicate a serious vibration problem. Noisy gearboxes, for example, may be the result of the gear frequencies exciting natural frequencies in the gearbox cover which in turn radiates the acoustic energy.

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Unusual Sources

There are many cases of unusual sources of vibration: foreign material or objects which move inside a rotor, water weeping in airfoil blades, entrapped water in a rotor which turns to steam when the rotor reaches operating temperature, axial clearance used up by thermal expansion, both bearings locked on a shaft, uneven stress caused by dissimilar material, electrical discharge through bearings, etc.

Suspect the unusual when the levels are very high, when phase and amplitude readings do not repeat from one run to the next, or when the readings are erratic.

- Tennis ball inside automobile tire
- Can of baked beans inside intake manifold
- Clean-up man dumping cold water in fan intake

Background Vibration

The background vibration levels should be checked with the machine shut down. Much time has been spent trying to determine the machine cause of a frequency in the spectrum which later turned out to be present in the background and not part of the machine under study.

If both the background and the machine vibration are repetitive, the background levels can be removed from the spectrum by linear averaging while the machine is running and negative averaging with the machine shut off.

Summary

Remember, vibration analysis is not always as simple as we would like it to be. However, when problems are approached logically, recognizing what is known about the machine in question and its history, the solutions to the vibration problems can be found.

Determine the turning speed frequency. Then the frequencies present in the spectrum can be categorized as subsynchronous, synchronous and nonsynchronous energy.

Don’t give up during the analysis process and remember to think about the possible unusual causes.

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Section Two
Review of Vibration Basics

Objectives

This section provides a review of the basics of vibration theory.

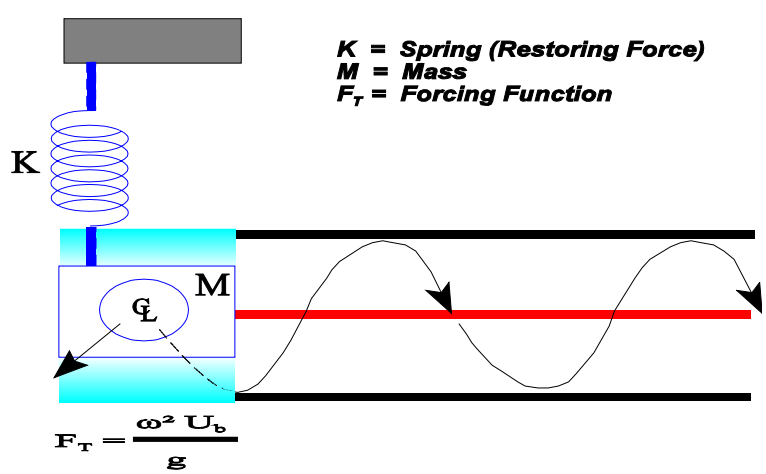
- Describe the fundamentals of vibration theory covered in Vibration Analysis I or other basic vibration courses.
- Apply this information as a foundation for later discussions on vibration theory and fault types.

Fundamentals of Vibration Theory

Vibration is the movement of a body about its reference position.

- The vibration occurs as a result of an excitation force that causes the motion.
- This exciting force may either be random or periodic.

The remainder of this chapter discusses forced periodic vibration such as vibration found in rotating equipment. A classic example of vibration is a weight supported by a spring. The weight has mass (**M**), and the spring has stiffness (**K**). If a forcing function ($\mathbf{F_T}$) such as unbalance is present, the mass will move back and forth across its reference or starting position.



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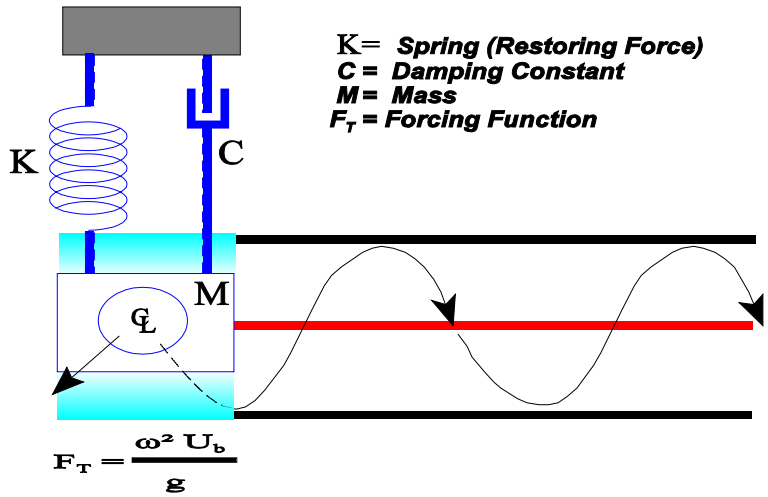
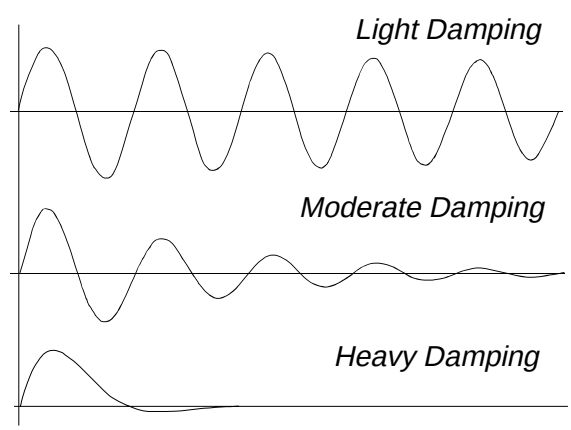
As long as there is a forcing function, the mass will continue to vibrate forever. The diagram below shows a damper that adds resistance to the motion. Some systems have dampers to isolate the vibration from other structures and machines.



System Damping Examples



The amount of damping actually determines the resistance the system will have to motion. As seen above, a lightly damped system offers little resistance to motion; therefore, it takes a longer time for the signal to decay. Most rotating mechanical systems have very little damping. As long as the forcing function (unbalance, misalignment, gear/bearing impacts) remains, the vibration will continue. This vibration is repetitive and lends itself well to **FFT** (Fast Fourier Transform).

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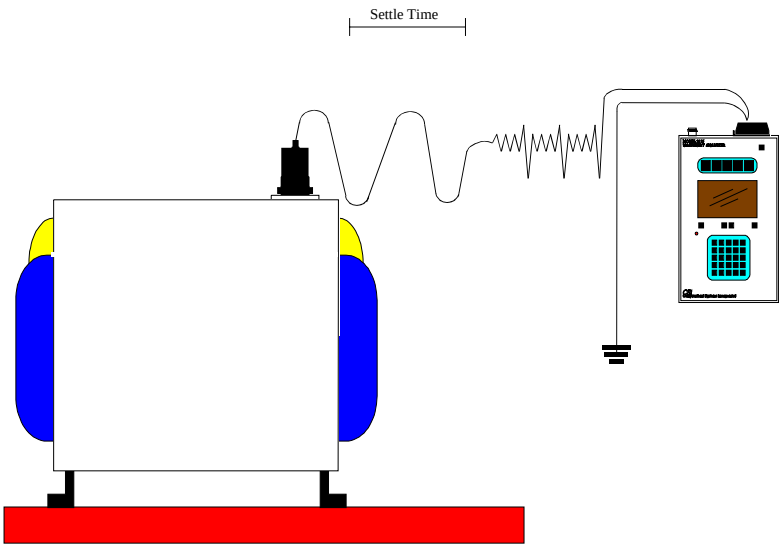
Vibration Transducers

In order to measure and analyze a machine's vibration signature, the mechanical vibration must be converted into a voltage signal that can be analyzed by the spectrum analyzer. The amplitude of the voltage signal varies with time, which causes the time waveform on which the FFT is performed to arrive at the spectrum.



A device called a ***transducer*** is needed to convert the mechanical vibration to the electrical signal. In the world of vibration analysis, there are three primary groups of transducers.

- Displacement Probes
- Seismic Velocity Probes
- Accelerometers

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Displacement Probes

Displacement probes are noncontact probes that measure the relative motion between the probe tip and the vibrating element. They convert mechanical vibration into a time-varying voltage proportional to the displacement of the vibrating surface. Usually displacement probes are permanently or semi-permanently installed on the machinery. These probes have a limited frequency range of about 1,000 Hertz.

Seismic Velocity Probes

Seismic velocity probes are not as commonly used in the vibration industry as they once were. They are composed of a permanent magnet spring mounted inside a coiled wire. These probes convert the mechanical vibration into a time-varying voltage proportional to the velocity of the vibration. Due to the magnet moving relative to the coil, the probe is self-generated and does not require external power. These probes also have a limited frequency range of about 1,000 to 2,000 Hertz.

Accelerometers

Accelerometers come in various types. One of the most common is the piezoelectric accelerometer. These probes contain a piezoelectric crystal that emits a small voltage proportional to the acceleration levels sensed by the transducer. Accelerometers are available in many different designs with widely varying frequencies. Special accelerometers are available that convert acceleration to velocity before the voltage signal reaches the transducer cable.

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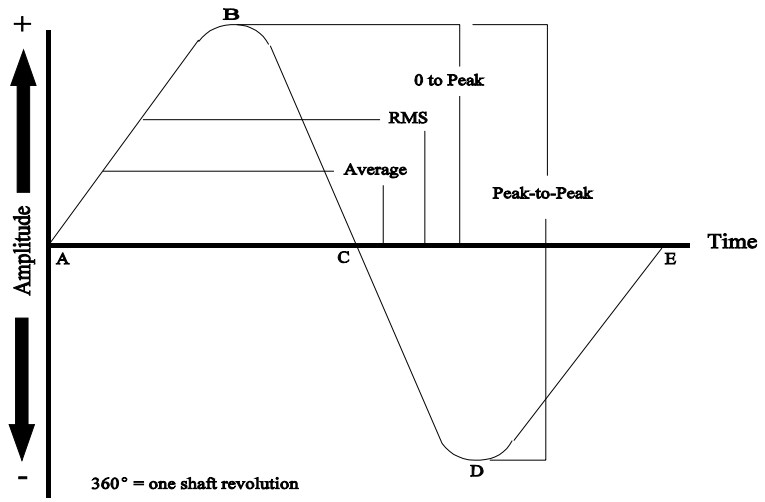
Mounting Resonances

Mounting resonances occur more often in accelerometers with higher frequency ranges available. A ***resonance*** is the frequency at which a structure's vibration is amplified due to structural considerations — low levels of damping or low stiffness. Transducer mounting resonances are affected by how the sensor is attached to the machine or vibrating surface. A very stiff method of attachment, such as a stud mount, generally moves the mounting resonance to a frequency above the frequency range of interest.

It is important when selecting a transducer to select the appropriate transducer for your application and an attachment method that will place the mounting resonance out of the frequency range of interest. Transducer selection and mounting are covered later in the section on Transducer Selection.

Amplitude Units



A trace of the displacement over time reveals a sinusoidal waveform pattern. The amount of vibration, the ***amplitude***, varies depending on which units are used to describe the vibration (displacement, velocity or acceleration).

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The typical units for each of these amplitudes are:

Displacement	**=**	**Mils, Peak-to-Peak**
Velocity	**=**	**Inches/sec, Peak**
Acceleration	**=**	**g's, RMS (1g = 386 inches/sec/sec)**

For most PDM applications, vibration data displays in units of velocity, which treats high frequencies and low frequencies equally. Acceleration enhances the amplitudes of high frequency data, and displacement enhances the amplitudes of low-frequency data.

2120 Lab - Analyze/Monitor Mode

❶ Connect 2120 to communications cable.

❷ Start MasterTrend.

❸ Open Database **INTMDVIB.DAT**.

❹ Use Analyzer communications to download a route from the station **DIAGNOSTIC LAB 1** named **INT. VIB EXMPL**.

❺ Connect lab table BNC X output to model 728 voltage adapter ***volt*** input (in this lab, the lab provides power).

❻ Monitor variable speed rotor per instructor's directions on following pages.

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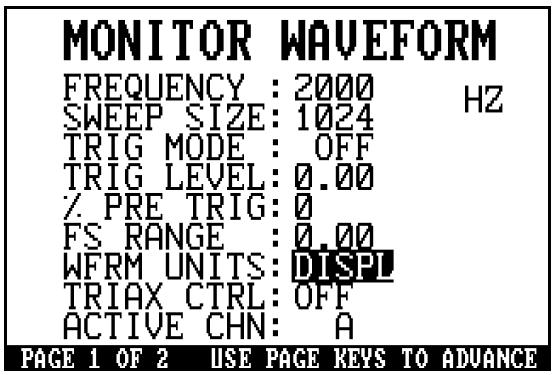
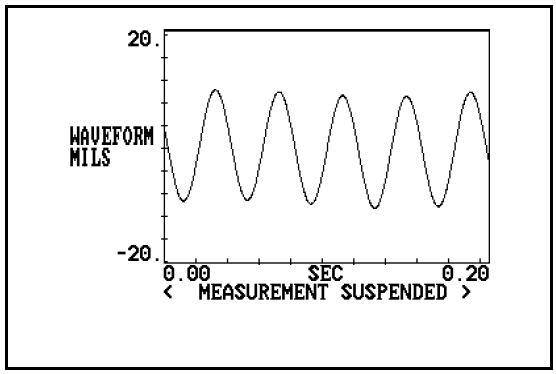
Displacement



Using the **Analyze, Monitor Mode, Monitor Waveform,** set up the analyzer to monitor the signal from the variable speed rotor.



Let's look at the simple rotor kit running at around 25 Hz (1500 RPM) with some unbalance as the predominant forcing function. We know from earlier studies that an unbalance force creates a 1x turning speed peak in the spectrum. We will start by looking at the ***displacement of the rotor with respect to time (D)***, — a digital recreation of the movement from the at rest position (centerline) over time.

The distance traveled by the rotating mass can be described by measuring the maximum positive displacement and the maximum negative displacement — the ***Peak-to-Peak Displacement***. This is normally measured in mils (1 mil = .001"). This could be measured mechanically with a dial indicator at slow speeds, but it becomes impossible at higher speeds. A more convenient way is to use an accelerometer to measure the acceleration and mathematically ***integrate*** to obtain displacement.

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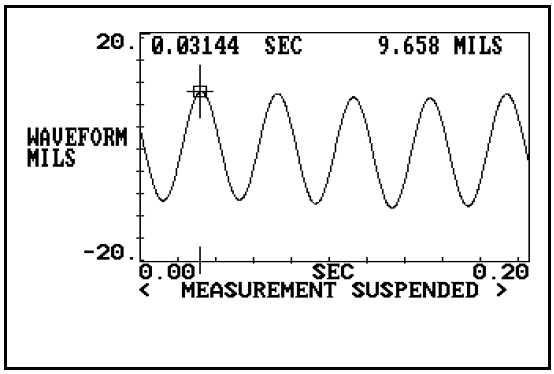
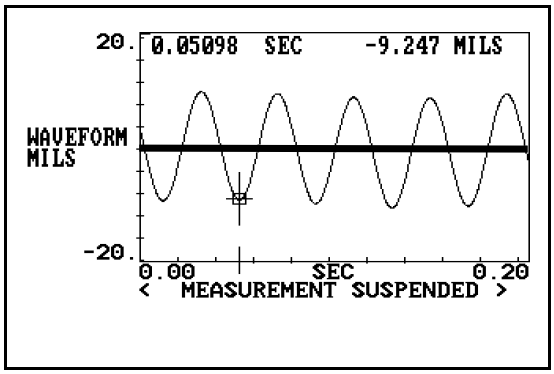
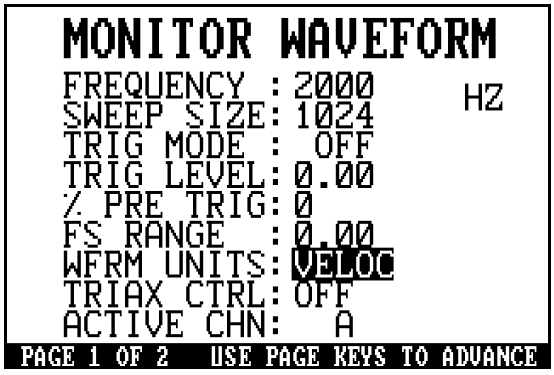


The cursor on the waveform above marks the **peak positive** value of the time waveform. The value is approximately 9.6 mils peak (your values may differ). Industry standard describes displacement in **Mils Peak-Peak**. To get this value, we need to find the **peak negative** value.



The cursor marks the **peak negative** value at approximately -9.2 mils peak. The value of this time waveform is **18.8 mils pk-pk**.

Velocity



Change the **Waveform Units** to **Velocity**.

To find the speed at which the unbalance mass travels, divide the distance traveled (displacement) by the time it took to travel this distance. This speed is the ***velocity*** (V). Inches per second (in/sec) is the industry standard unit.

$$\Delta D / \Delta T = \textit{Velocity}$$

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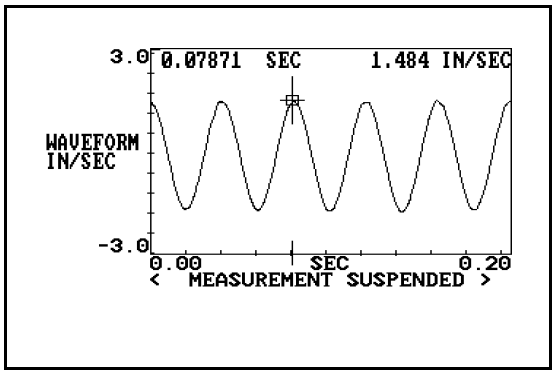
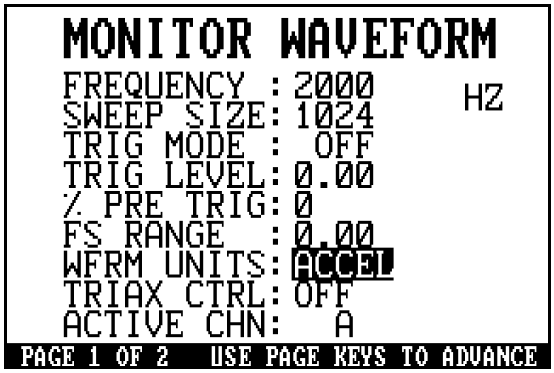


Velocity is normally measured **0-Peak** or simply **Peak**. In the example above, the value is 1.484 in/sec.

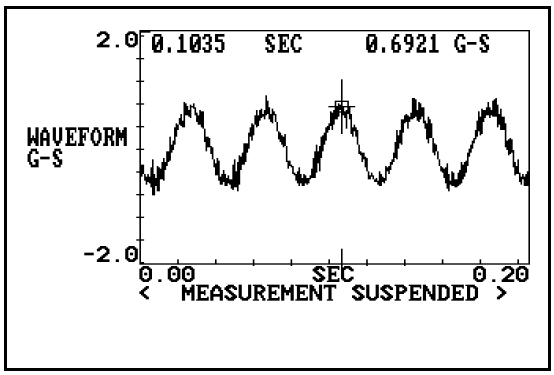
Acceleration



Change the **Waveform Units** to **Acceleration**.

Circularly the mass is moving at a constant velocity. In the vertical direction — what is being measured — the unbalance mass stops and changes direction at the top and the bottom. Vertically, the velocity changes during one complete cycle of motion. The rate that the velocity changes is called the ***acceleration*** (A).

$$\Delta V / \Delta T = \textit{Acceleration}$$



Acceleration is normally measured in **g's RMS**. In this example, the peak value of 0.6921 is multiplied by .707 to obtain the value in g's RMS units.

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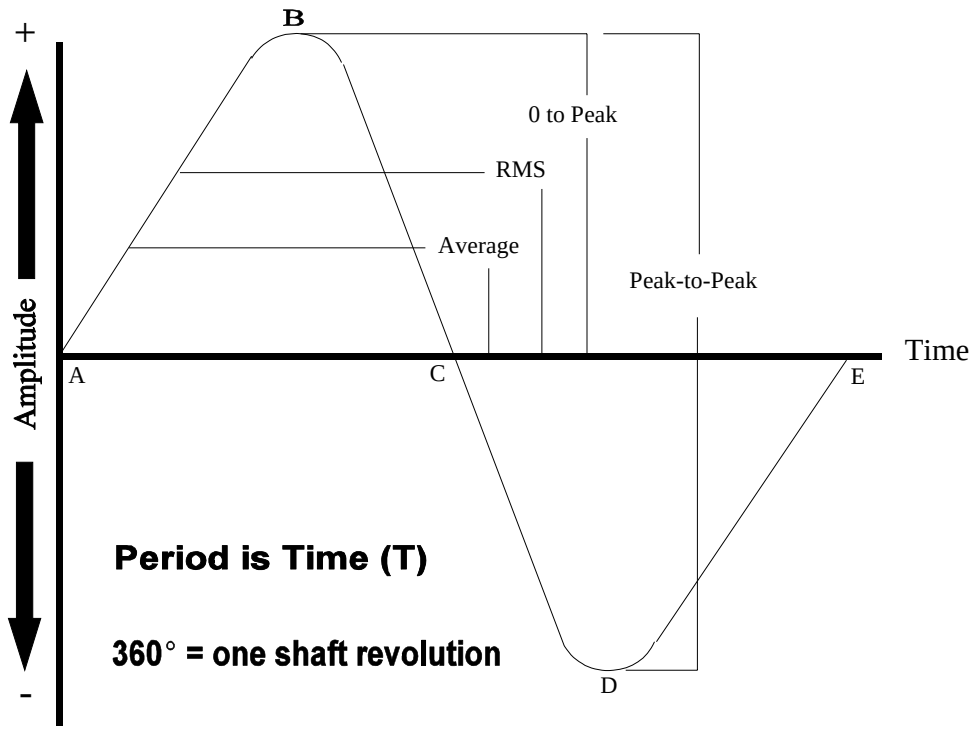
Time Units

Let's look at the basics describing the units of the time waveform again. This time, instead of the amplitude units (Y axis) we will look at the time units (X axis).



The time from **A** to **E** is how long it takes the event to happen. We can measure this time and relate it to a ***frequency (F)***.

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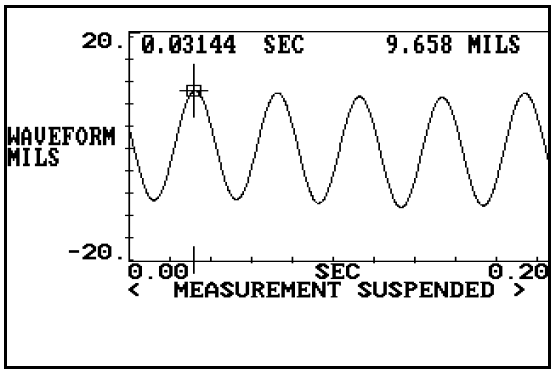

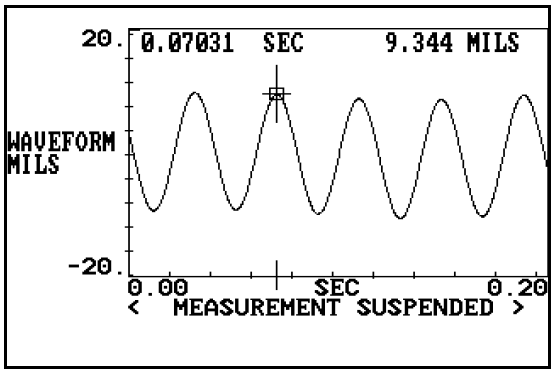

We can see that at time 0.03144, the unbalance force was at its peak positive position. At time 0.07031, the mass has completed the cycle. This time is referred to as ***T*** or ***Period***. Since the time it takes for the unbalanced mass to return to its starting position is also equal to one revolution, we can calculate the time it takes by subtracting the two values.

$$\Delta T = .07031 \text{ sec.} - .03144 \text{ sec.}$$

$$T = .03887$$

The following equation links event times in ***seconds*** to ***frequencies in Hertz***:

$$F \ (Frequency) = \frac{1}{T \ (Time)}$$

$$Frequency = \frac{1}{Time \ in \ Seconds} = \frac{1}{0.03887} = 25.73 \ Hz = 1543 \ RPM$$

This calculation is not very difficult for the given example, but a typical vibration signal has a very complex time waveform with many different signals combined into one. We cannot easily pick out frequencies or amplitudes so we must convert the time waveform into a more usable format. The mathematical process of converting from the time domain to the frequency domain is called the ***Fast Fourier Transform (*FFT)**.

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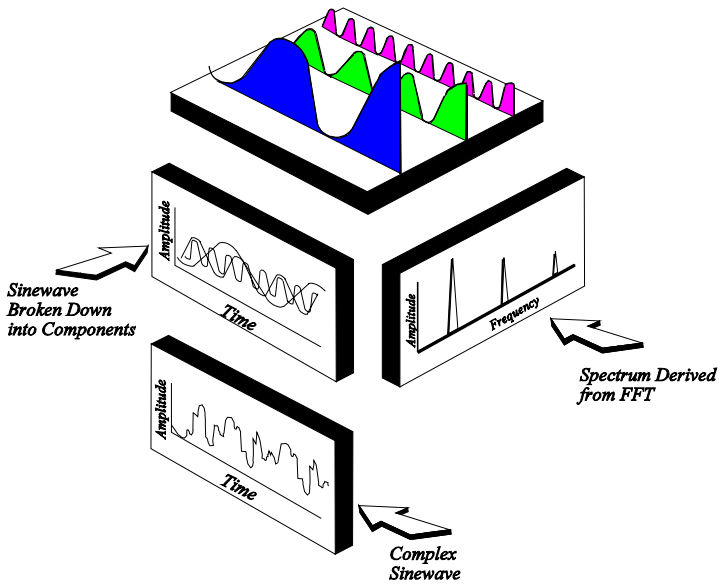
Frequency Spectrum

The Fast Fourier Transform (FFT) converts time waveform information to frequency-based information. The FFT has the graphical affect of rotating the waveform 90° so that the waveform is coming out of the page toward the reader. Many vibration faults are more easily diagnosed in the frequency spectrum than in the time waveform, because the frequency spectrum can separate closely-spaced frequencies and allow for the viewing of relative amplitudes.



In order to get full benefit from a spectrum, the analyst must understand signal processing, bandwidth calculations, lines of resolution and transducer selection. These topics will be covered later in this manual.

In review, the frequency units used to identify frequencies in a spectrum are:

Cycle per Minute = CPM
Cycle per Second = Hertz (HZ)
Cycle per Revolution = Order

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2120 Lab - Frequency Spectrum

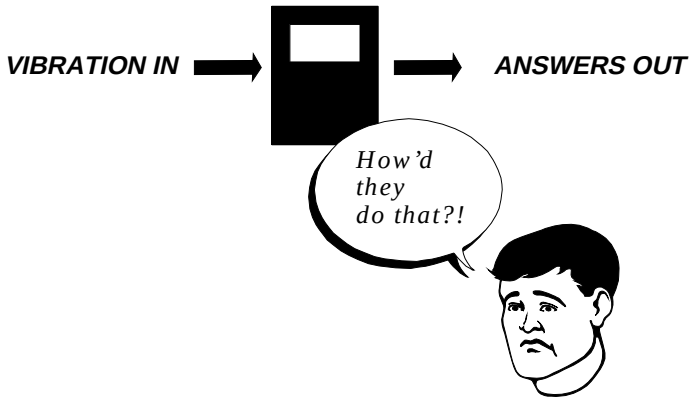
❶ Collect data from variable speed rotor kit (IMBAL EX) for the points labeled **VSD, VSV, VSA**.

❷ Dump the data to MasterTrend for viewing.

Machinery faults occur at specific frequencies that are often difficult to recognize in the time waveform. This makes the spectrum a very useful tool. The time waveform, although not as useful for fault detection, is valuable for determining the character of the vibration such as impacting symmetry.

Summary

This section reviewed some basic vibration concepts. The main points to remember are the definitions of time waveform, frequency spectrum, amplitude units, and frequency units. It is also important to know the various types of vibration transducers.



The goal of this course is to remove the black box mystique from the vibration measurement process. It is important for an advanced vibration analyst to understand the material in this manual and the analysis processes behind many of the choices made on a daily basis in the vibration industry.

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Section Three
Effects of Amplitude and Phase

Objectives

- Describe the effects of amplitude units on frequency and time domain data.
- Explain what phase is.
- Explain the phase shifts associated with amplitude units.
- Discuss the importance of transducer polarity.

Amplitude Unit Characteristics

Effects in the Spectrum

Basic vibration training introduced three units used to measure amplitude. Velocity, Displacement, and Acceleration apply to amplitude in both the frequency domain and the time domain. The most common amplitude unit for vibration trending is Velocity. Displacement may be used when relative motion or slow speed is a consideration. Acceleration is often used in gearbox and high speed machinery as well as bearing troubleshooting.

Effects in the Time Domain

As expected, the amplitude units affect the time domain much in the same manner as the frequency domain. The next few examples show the effects of amplitude in both the spectrum and waveform.

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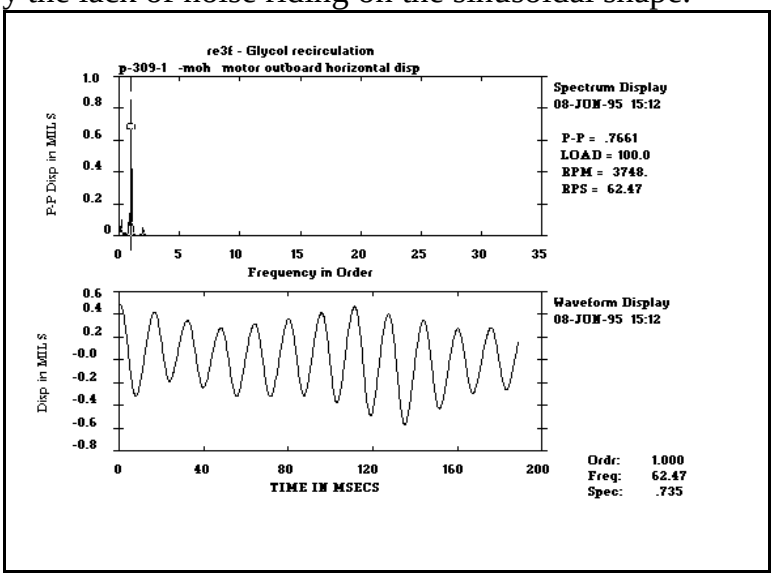
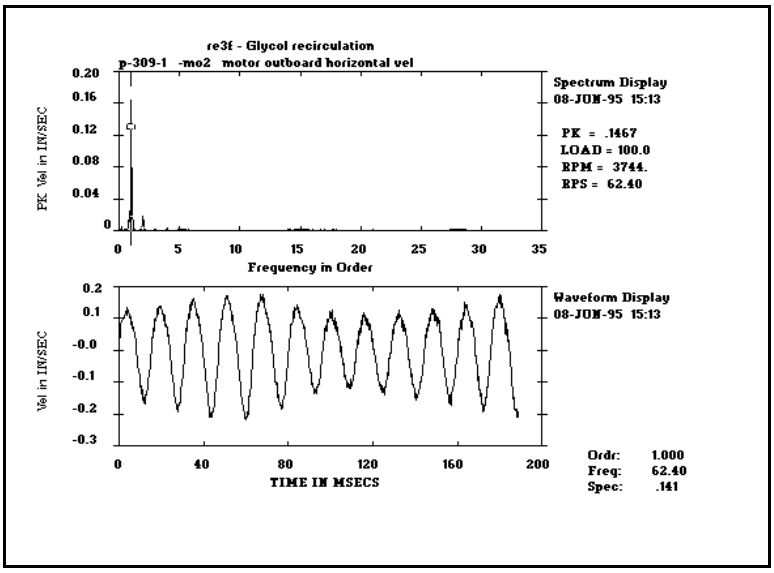
Displacement

The spectrum and waveform display the attenuation of the high frequencies. The spectral plot clearly shows no high frequency data. This is evident in the waveform by the lack of noise riding on the sinusoidal shape.



Velocity

Viewing data linearly across the frequency range is a characteristic of Velocity. Some energy is present around 15xTS and 30xTS in the spectrum. This energy is reflected in the waveform as the noise riding in on the sinusoidal shape. This high-frequency energy was not present in the displacement waveform. Again, the dominant shape is sinusoidal due to the characteristics of velocity.



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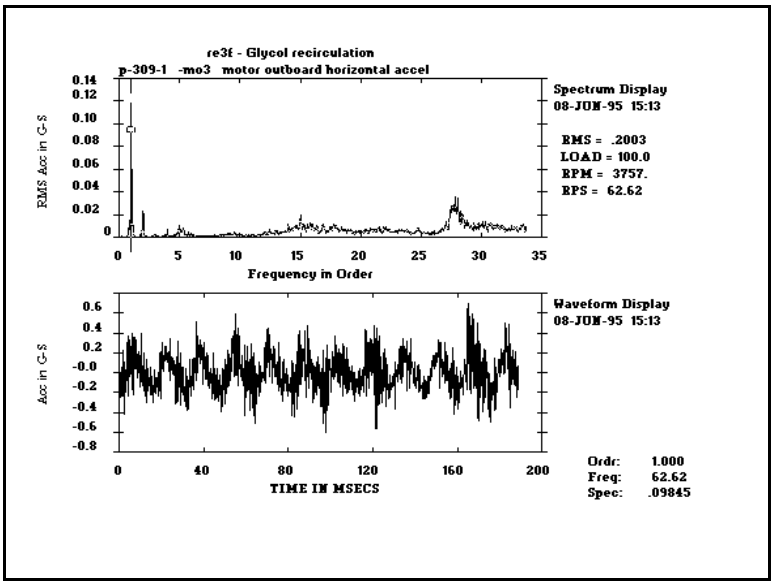
Acceleration

Accentuating high frequencies and attenuation of low frequencies are characteristic of Acceleration. Notice the raised noise floor in the spectrum beginning at approximately 12xTS. The waveform has an underlying sinusoidal shape, but the high frequencies are very prominent. These amplitude units also affect the relationship of phase data.



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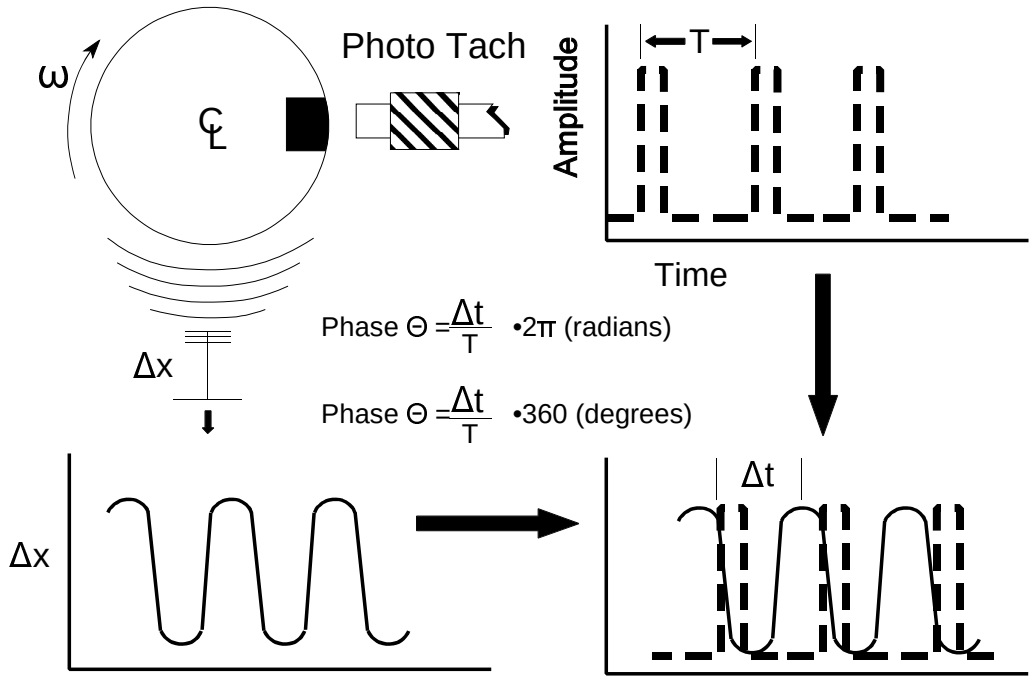
Phase Relationships in the Time Domain

What is Phase?



Single-Channel Phase Measurement (@ 1XTS)

Phase is one of the most important single-channel analysis tools for a vibration analyst. ***Phase*** is a relationship — simply the difference in time between two events. We talk about two different types of phase in vibration terms — ***absolute phase*** which is really the relationship of the peak vibration as it relates to some reference signal and ***relative phase***. The most common frequencies for single-channel phase analysis are 1x and 2x RPM signals. The peak of this signal is compared to a reference mark (reflective tape, keyway, stroboscope). The time relationship of the reference mark and the peak vibration is converted to an angular value from 0 to 360° (one revolution). In later chapters we will look at relative phase relationships between two vibration points using the two-channel capabilities of the 2120 (Cross Channel Phase). The analyst uses phase when trying to balance an imbalanced rotor to locate the heavy spot. Phase is also a useful tool to determine types of unbalance, misalignment, looseness, soft foot, bearing misalignment and resonance which we will look at in the appropriate chapters.

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2120 Lab - Peak Phase

Press **Analyze**, select **6) MONITOR MODE**, and then select **5) MONITOR PEAK/PHASE**.

```
ANALYZER FUNCTIONS

 1) DISPLAY SPECTRUM
 2) DISPLAY WAVEFORM
 3) DISPLAY PARAMETRS
 4) STORE DATA
 5) ACQUIRE SPECTRUM
 6) MONITOR MODE
 7) ZOOM ANALYSIS
```

```
MONITOR MODE

 1) MONITOR SPECTRUM
 2) MONITOR WAVEFORM
 3) MONITOR OVERALL
 4) MONITOR TEMPERATR
 5) MONITOR PEAK/PHAS
 6) MONITOR DC
 7) MONITOR BATTERY
```

We will monitor **1X RPM** or **1 Order**. The defaults will be good for the rest of the choices.

```
MONITOR PEAK/PHASE

ORDER (XRPM)    :1
BANDWIDTH(XRPM):0.10
AVERAGER ENABLD: NO
MINIMUM RPM     :0
TACHMTR TIMEOUT: NO
FS RANGE        :0.00

PRESS EITHER PAGE KEY FOR STORED DATA
```

```
MONITOR PEAK/PHASE

RPM  :  1450.
PEAK :   1.48  IN/SEC
PHASE:     88 DEGREES

[PRESS CLEAR TO CLEAR DATA]
[PRESS ENTER TO EXIT SCAN ]
```

Notice the phase reading. We will now show how the phase is calculated. Press the **Reset** button.

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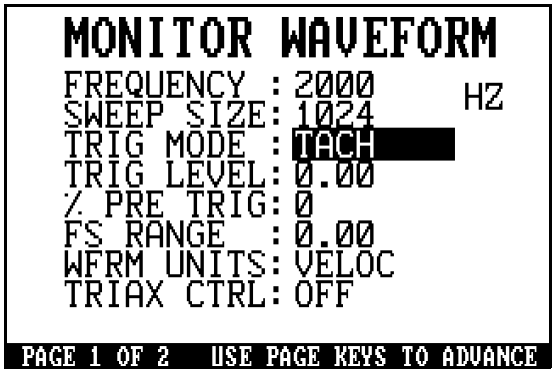
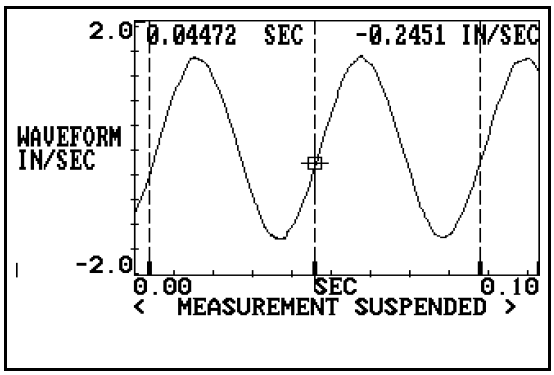
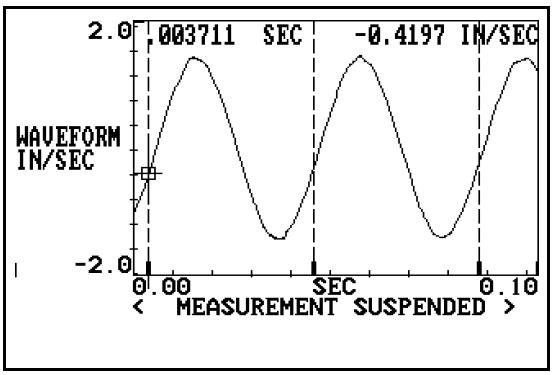
Let's take a look at how this Peak/Phase reading is developed. Select the **Analyze** function and then **MONITOR MODE** and **MONITOR WAVEFORM**. We will turn the **TRIGGER MODE** to **TACH**.



When you have a stable signal, press **Enter** to suspend the measurement.





Use the **Exp** and **Dec** keys to adjust the display as shown. We need to know the **total time for one revolution**. Use the left and right arrow keys to move the cursor until it lines up with the tach marker near the center of the screen. Write this time at **1.** below. Use the left arrow to cursor to the previous tach marker. Write the time at **2.** below. To calculate the time for one revolution (360°), subtract **2.** from **1.** to get **T**.

1. _____		Time 1
2. _____	-	Time 2
		Δ Time
T _____		

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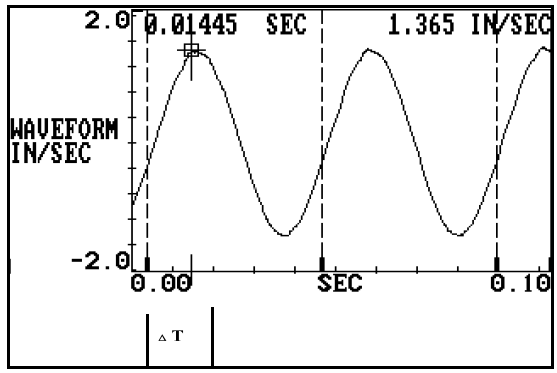
Next we need to find when the peak is reached. Use the arrows to move the cursor to the peak positive vibration. Write this time at **3.** below. We will use a proportion to calculate the time event relationship to 360°.



3. _____ Time 3
2. _____ - Time 2
_____ Δ Time

Your answer should be close to the reading you had monitoring the peak/phase.

$$\textit{Phase } \theta = \frac{\Delta t}{T} \cdot 2\pi \text{ (radians)}$$

$$\textit{Phase } \theta = \frac{\Delta t}{T} \cdot 360 \text{ (degrees)}$$

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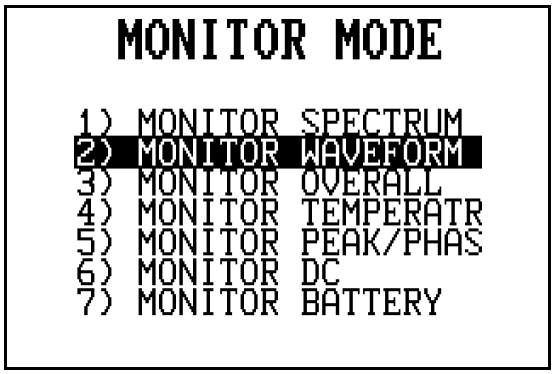
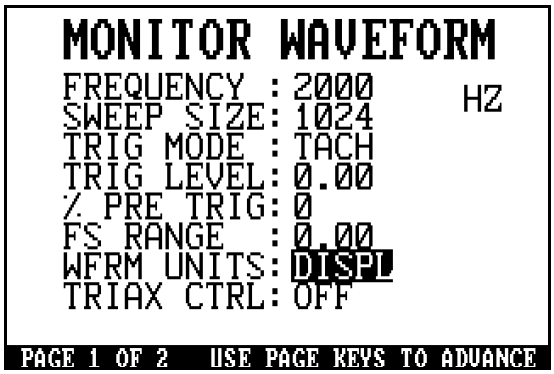
2120 Lab - Phase vs. Amplitude Units

We will look at our rotor kit time waveform signal again — this time with a tach signal. We have placed a piece of reflective tape on the rotor and an accelerometer on the unit.

Press the **Analyze** button and then **2) MONITOR WAVEFORM**. We will use the following setup:





Make sure **TRIG MODE** is set to **TACH**.

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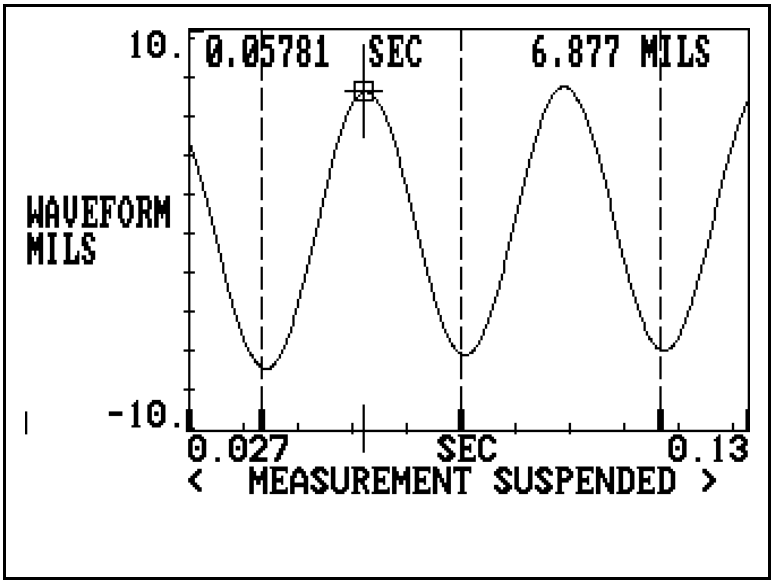
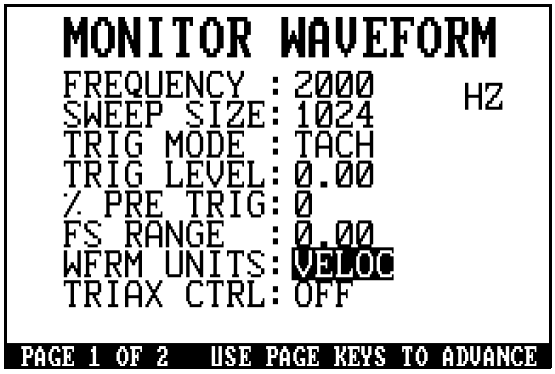
You may need to use the **Exp** and **Dec** keys to adjust the display for 2 to 3 revolutions (vertical dashed lines).



The displacement is at its peak about halfway between the two tach pulses. Remember the tach pulses represent one revolution or 360°. The peak displacement in the signal above is close to halfway between the two markers or approximately 180°.

Now change the waveform units to velocity. Press **Enter**.



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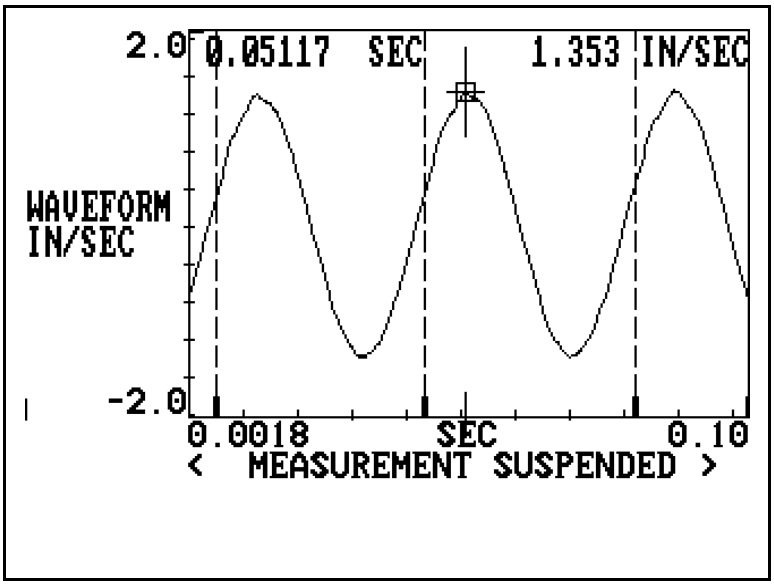
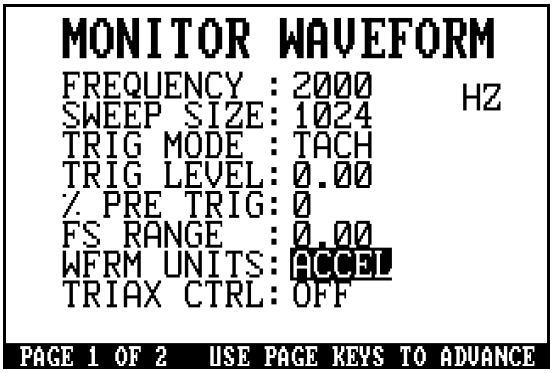
Again, adjust the **Exp** and **Dec** keys for the same display.



The peak velocity above occurs about 1/4 of the distance between the tach markers. This equal to approximately 90°.

Now change the waveform units to acceleration. Press the **Enter** key.



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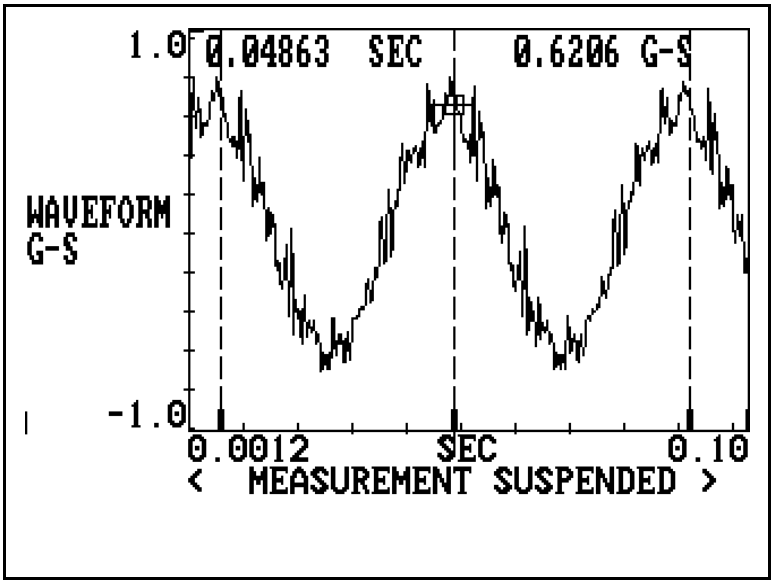
You may need to adjust the **Exp** and **Dec** keys again.



The peak acceleration takes place on the tach marker. This is equal to approximately 0°.

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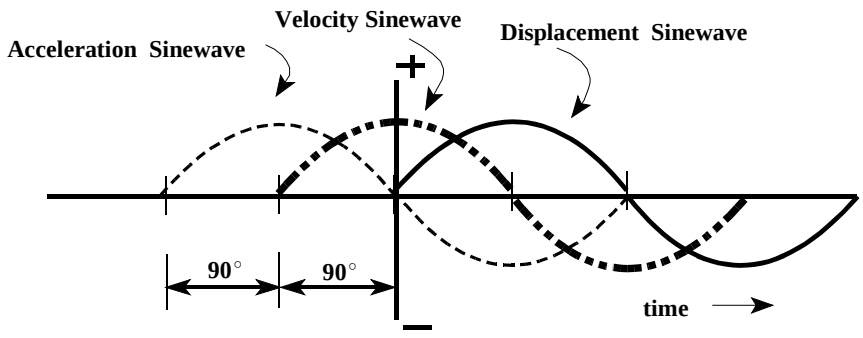
Phase vs. Amplitude Units

What we have just seen is that there is a significant difference between the phase relationships of the three different amplitude units — acceleration, velocity and displacement. The laws of physics and relationship of vectors govern this topic. The figure using displacement as the base unit type, illustrates that velocity leads displacement in phase by 90°. Also, acceleration leads velocity by 90° or leads displacement by 180°. This is important when comparing phase readings collected with accelerometers and noncontact displacement probes. The figure shows the phase shift caused by selecting different amplitude units.



The sinewaves above show the phase relationship between the three most typically used transducer types.

The selection of different amplitude units is just one source of hardware-induced phase shifts.

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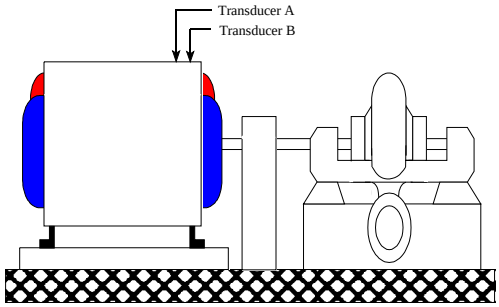
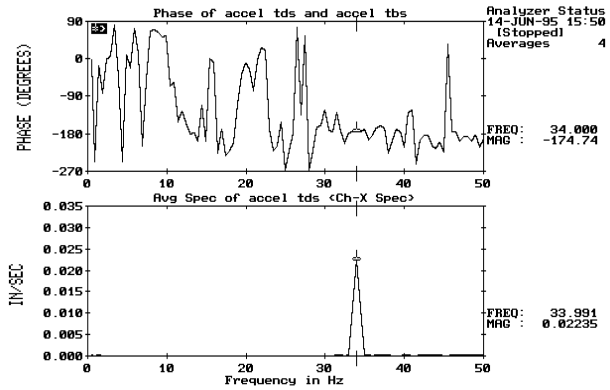
Transducer Polarity

An indication of a phase shift across a machine may or may not be caused by the machine. If the phase data was collected with a multichannel analyzer, then the issue of transducer polarity must be addressed. The internal wiring of vibration transducers will vary with model and manufacturer but can also vary between the same model. Therefore, a phase shift on a piece of equipment may be due to the transducers being used rather than a machine fault. For this reason, it is extremely important to check the polarity of any transducers used for phase collection.



The cross-channel phase data was collected from the machine pictured above. Both accelerometers were the same model number and mounted in the same orientation on the motor. The phase data being compared is associated with turning speed of the motor. The expected phase shift would be 5 or 10° between the transducers. Notice the actual phase shift is 174.74°. The induced phase shift between the transducers was caused by the internal connections being reversed. If this difference in polarity was not recognized prior to data collection, an incorrect analysis may result.



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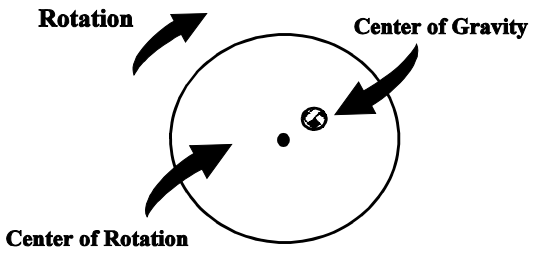
Section Four
Unbalance

Objectives

- Describe the spectral and waveform characteristics of unbalance.
- Identify the different types of unbalance using spectral data and phase data.
- Associate the special conditions of vertically mounted equipment with the balancing process.

What is Unbalance?

Unbalance, sometimes referred to as imbalance, is very simply the result of a ***shaft's center of mass not rotating at the center of rotation***. This results in a heavy spot on the rotor. This heavy spot produces a centrifugal force that forces the rotor to rotate off center and causes a high vibration amplitude at the 1xTS frequency in the spectral data and sinusoidal waveform in the time domain data. Phase data is used to locate the heavy spot on the rotor.



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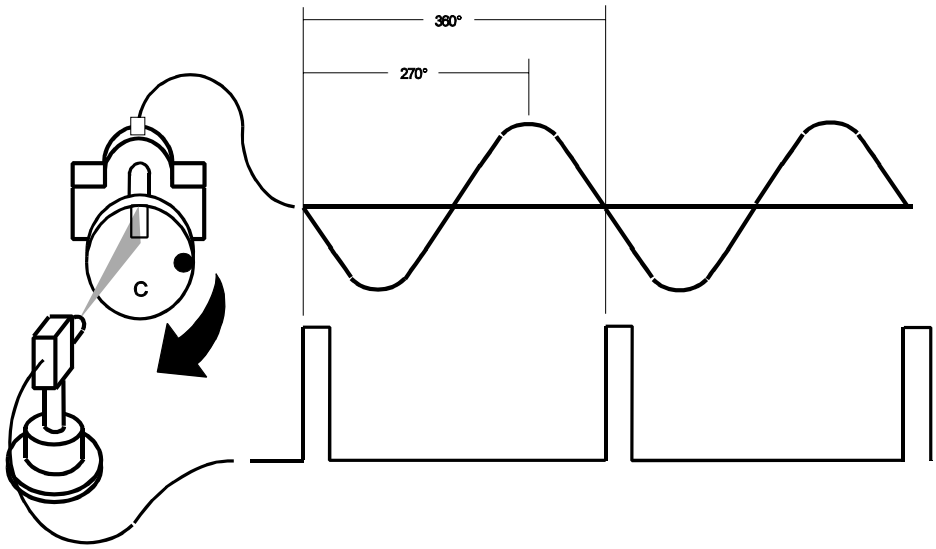
It is important to realize that a vibration sensor and a tachometer signal cannot directly identify the heavy spot on the rotor. The tach signal and the sensor actually identify the high spot. The high spot is defined as the measured response to the heavy spot on the rotor. This is the point where the measured vibration is the highest.



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Phase Lag

A ***phase lag*** angle exists between the actual heavy spot on the rotor and the measured high spot. Remember that CSI equipment measures true phase, which means that the phase angle increases opposite to the direction of rotation from the reference mark. A factor that affects the angle of the phase lag is the frequency of the first shaft critical frequency. If the turning speed is much less than the shaft critical frequency, the phase lag will be close to 0°. As the shaft speed approaches the shaft critical, the phase lag will approach 90°. At resonance the phase lag will be 90°. Above resonance, the phase will be above 90°. As the shaft speed increases significantly above the resonant frequency, the phase lag will approach 180°.

The lag angle exists because during the time required for the vibration due to the heavy spot to be transmitted through the machine to the vibration sensor, the rotor is still turning, allowing the phase angle to continue to increase. One way of determining the phase lag angle is to place a known amount of weight at a known angular location on the rotor. This is known as a trial weight. The lag angle can be determined from the resulting amplitude and phase data, which is the effect of the trial weight.

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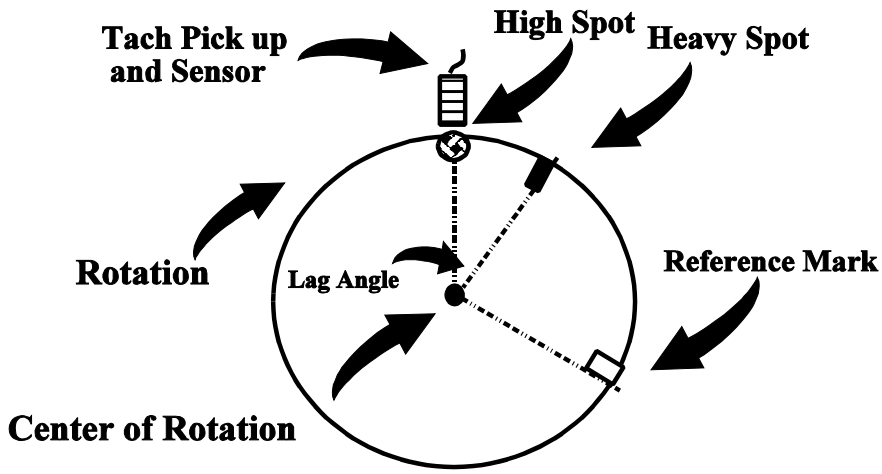

HEAVY SPOT is 90° past reference mark

HIGH SPOT is 120° past reference mark

LAG ANGLE is HIGH SPOT - HEAVY SPOT = 120° - 90° = 30° Lag Angle

Most computerized balancing programs will determine the lag angle during the balancing process. The concepts of heavy spot, high spot, and lag angle are important when balancing a rotor. The high 1xTS vibration occurs because the heavy spot passes the vibration sensor once per revolution, producing the measured vibration at the turning speed frequency.

FAULT GUIDE

Vibration	Dominant Frequency	Dominant Plane	Phase Reading
Unbalance			
Static	1xTS	Radial	Radial in phase
Dynamic	1xTS	Radial	Radial 0-180 out / 2 plane
Couple	1xTS	Radial/axial	Radial 180 out
Overhung rotor	1xTS	Radial/axial	Radial unsteady/Axial in phase

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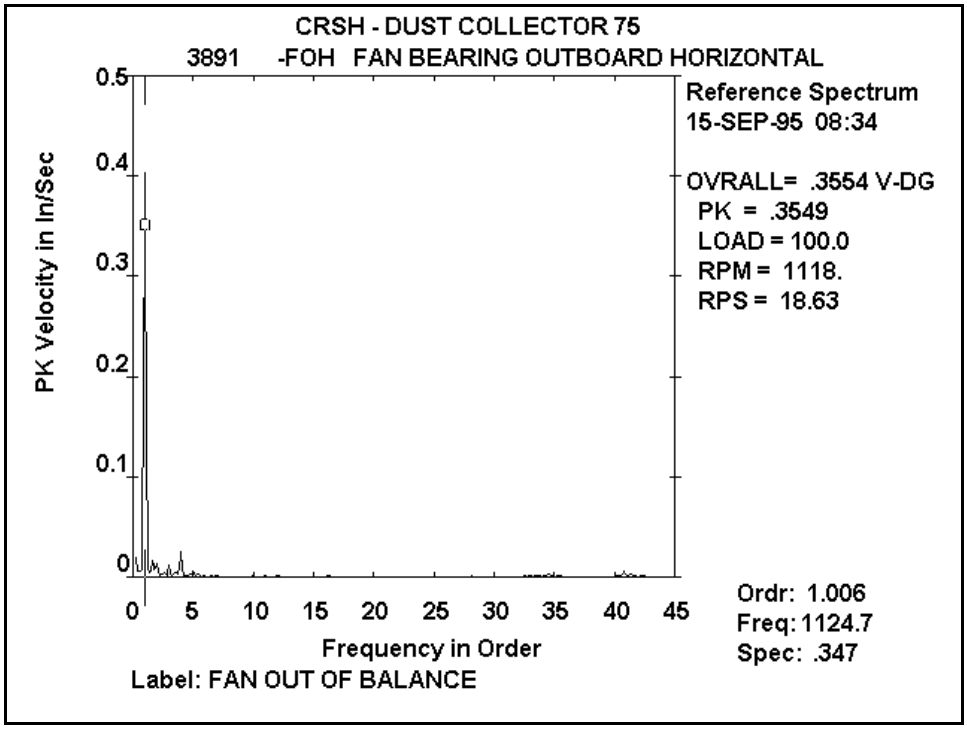
Spectral Data

When unbalance is the primary problem, a 1xTS will appear in the spectral data, and it will generally be at least 80 percent of the overall amplitude. The harmonics of 1xTS should be very low in amplitude, with low axial vibration levels. Also, if the turning speed frequency is below the first shaft resonant frequency, then the unbalance amplitude will increase proportionally with the square of the speed change. For example, if the speed doubles from 1000 RPM to 2000 RPM, then the 1xTS amplitude can be expected to increase by a factor of four.



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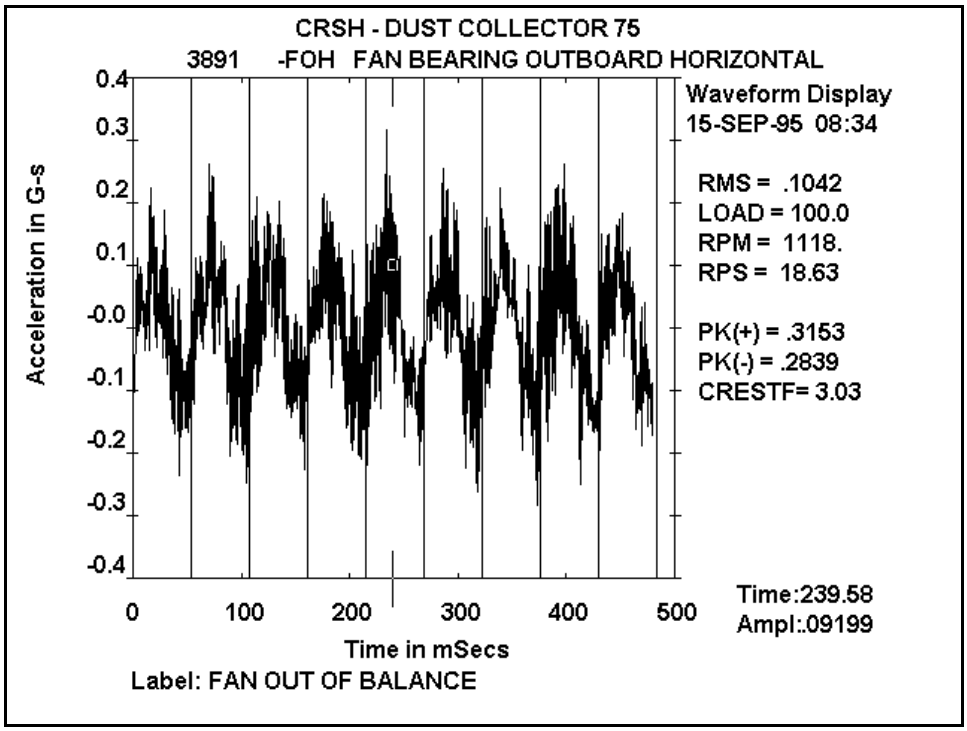
Waveform Data

Since the spectral data comes from the time waveform, we can expect the time waveform to have a strong sinusoidal pattern. The display units of the time waveform will affect the clarity of the sine wave. A displacement waveform will show a clean sine wave while an acceleration waveform will show a lot of high-frequency data riding on top of the sine wave.



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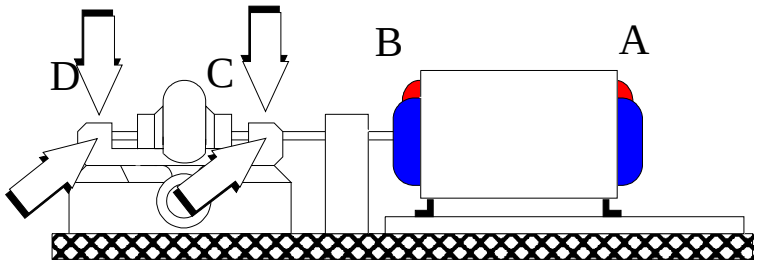
Types of Unbalance

Unbalance can be divided into two primary types — static and dynamic.

Static Unbalance

Static unbalance is often referred to as force unbalance. A simple force unbalance can be corrected with a single-plane balance correction. In a frictionless stand, the heavy spot will roll to the bottom and can be corrected with one weight 180° from the heavy spot. A statically balanced rotor should stay at any position in a frictionless stand.

The radial vibration readings are the highest amplitude with the axial vibration generally much lower in amplitude. Phase measurements can provide the data needed to identify static unbalance. Static unbalance will show a 0° phase shift across the rotor (vertical to vertical or horizontal to horizontal) and a 90° phase shift from vertical to horizontal at the same bearing location. Also, the measured phase angle will change the same amount that the heavy spot changes, if the system is linear.



90 °
Horizontal to Vertical

If a 0° or a 180° phase shift exists from the vertical to the horizontal measurement direction, then the machine is most likely running at resonance or is misaligned. The resonance or misalignment will need to be corrected before attempting to balance.

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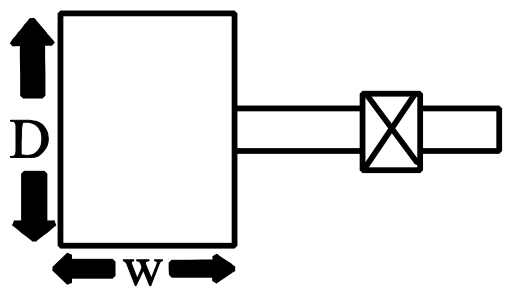
Dynamic Unbalance

Dynamic unbalance is actually every case other than static unbalance. This includes couple unbalance, overhung rotors, and vertically mounted machines. Dynamic unbalance will require two or more correction weight planes to balance the machine. A good rule of thumb to follow is that for a single rotor, two-plane balancing may be required if the rotor diameter, D, is less than four times the rotor width, W.

If D / W < 4, then two-plane balance is required.



D/w < 4

The vibration data for a dynamically unbalanced rotor will not differ significantly from static unbalance with the exception of the phase data. The high 1xTS radial vibration will still dominate the spectral data. The phase data is controlled by the location of the heavy spots on the rotor. To understand how the phase will differ from static to dynamic unbalance, we need to understand how the location of the heavy spots vary from static to dynamic unbalance.

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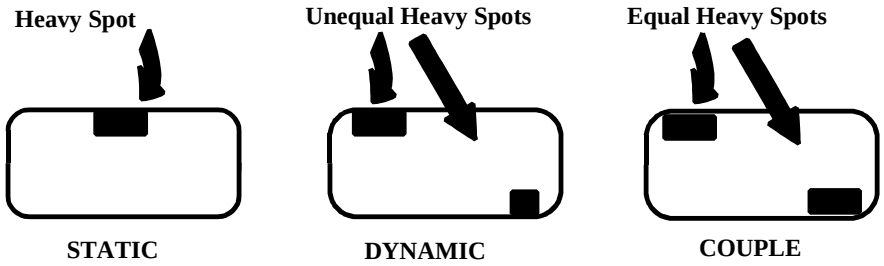
Static vs Dynamic vs Couple Unbalance

Dynamic unbalance is the result of:

- Two unequal heavy spots 180° apart in separate planes on the same rotor. Phase is likely to be 180° across the rotor, but the amplitudes will differ since the amount of the heavy spots differ.

- Two equal or unequal heavy spots in separate planes on the same rotor located at some spacing other than 180° apart. The amplitude and phase will be related to the location and the amounts of the heavy spots.

- If the heavy spots are equal and 180° apart, then the rotor will be dynamically unbalanced but statically balanced. The amplitudes across the rotor should be similar and the measured phase readings will be 180° apart. This special case of dynamic unbalance is called a ***couple unbalance***. Remember that although this rotor will stay at any position in a frictionless balance stand and is statically balanced, each end is unbalanced by the same amount, 180° apart. This requires correction in two planes.



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Other Considerations

Overhung Rotor

This subject always seems to get treated separately. However, if an overhung rotor is statically unbalanced, then it could be balanced in one plane. If it is dynamically unbalanced, then two or more planes will be necessary to correct the unbalance. The vibration will be high in the radial directions as seen with center hung rotors, but over hung rotors often show a high 1xTS axial vibration due to axial thrusting resulting from the unbalance.

The phase readings will provide the needed information to determine whether a single or multiplane solution is needed. If the phase readings from each bearing are the same (vertical to vertical, horizontal to horizontal) then the rotor could be balanced in one plane. This is provided the analyst has access to the center line of the rotor. If the rotor diameter is less than four times the rotor width, then two-plane balancing may be required.

Vertical Equipment

Vertical equipment really can be tricky because the unbalanced rotor is sometimes inaccessible. Usually, weights can be placed in a plane at the top of the motor so that a centrifugal force is created that pulls, counteracting the vibration at the top of the motor.

Summary

Unbalance is the simplest defect to identify in a vibration spectrum provided the turning speed of the machine is known. Be aware that, although unbalance shows up at 1xTS, many other defects also appear at 1xTS. This is why checking the phase can be helpful when confirming unbalance.

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2120 Peak/Phase Balance Lab

In this lab, we will use the **Analyze** function and **Monitor Peak/Phase** to evaluate an out of balance rotor.

❶ Setup the analyzer and associated equipment to monitor the rotor kit.

❷ Evaluate spectral data for unbalance.

❸ Monitor and record Peak/Phase data.

❹ Evaluate the results.

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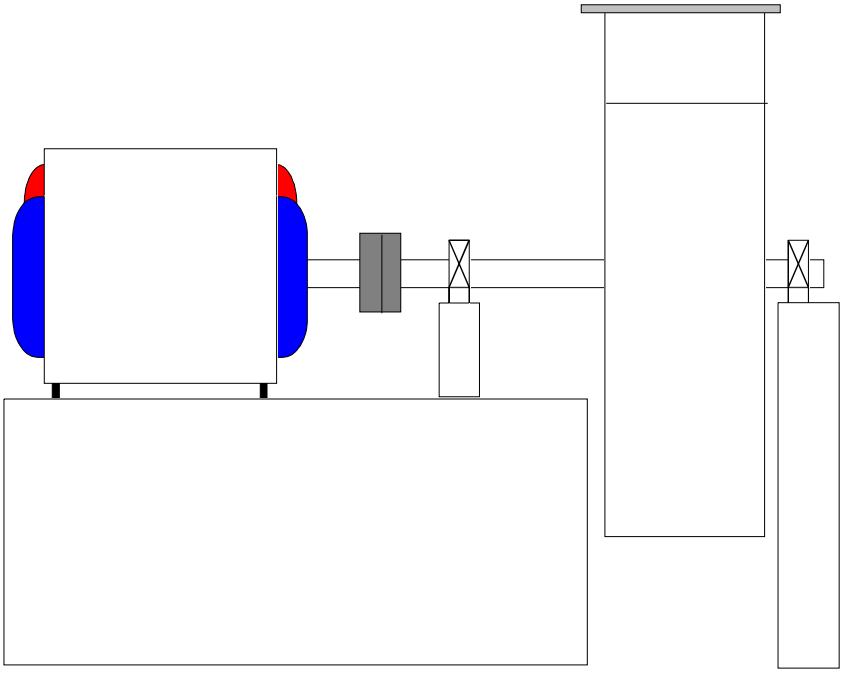
Case Histories - Unbalance

Unbalance Case #1 - Vent Blower



❶ 450-HP, 3600-RPM Motor.

❷ Fan is direct drive, center hung.

❸ Fan has 16 reverse curved blades.

❹ Material buildup has been a problem.

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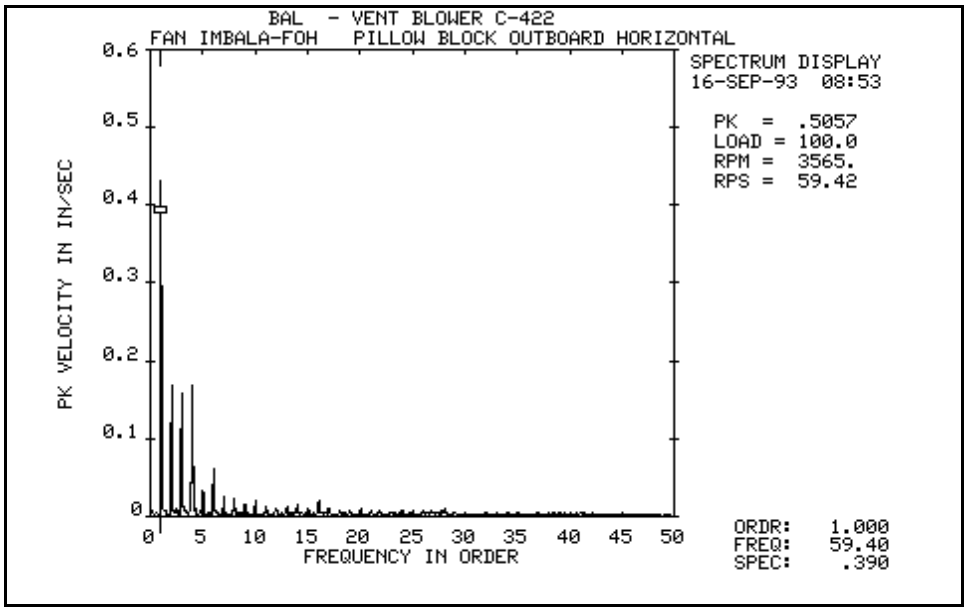

The spectrum above is from the Fan Outboard Horizontal (FOH) position. Notice the 1xTS peak with some harmonics of turning speed. The amplitude of the 1xTS vibration had exceeded the warning level on the monitoring panel. What problem should be corrected first, the unbalance or the looseness?

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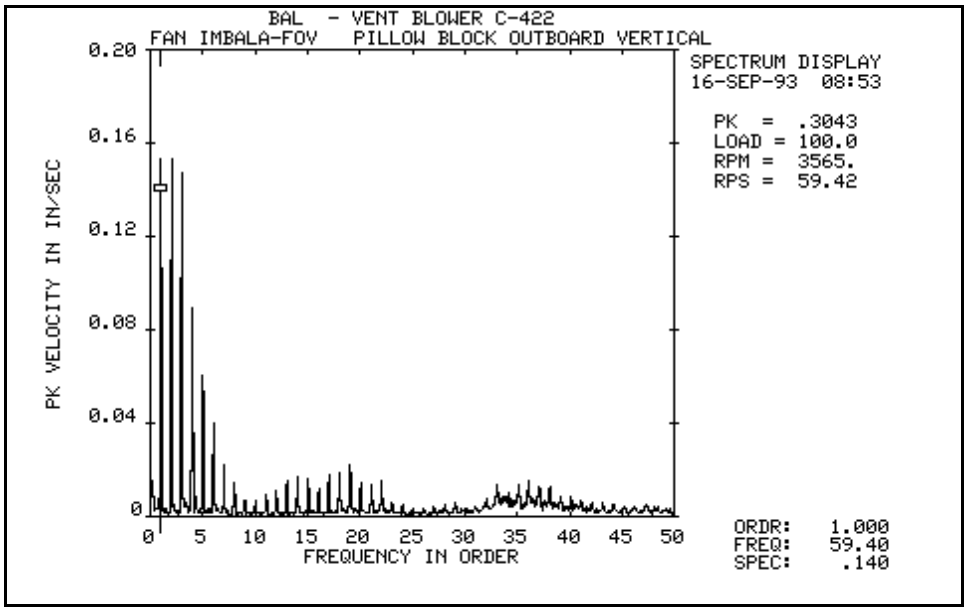

The Fan Outboard Vertical (FOV) plot is shown above. As expected, the 1xTS amplitude is lower than the FOH amplitude. Again, looseness is present in the data. It was determined that the looseness was being driven by the unbalance in the fan. The next page shows measurement points FOH and FOV after the fan was balanced.

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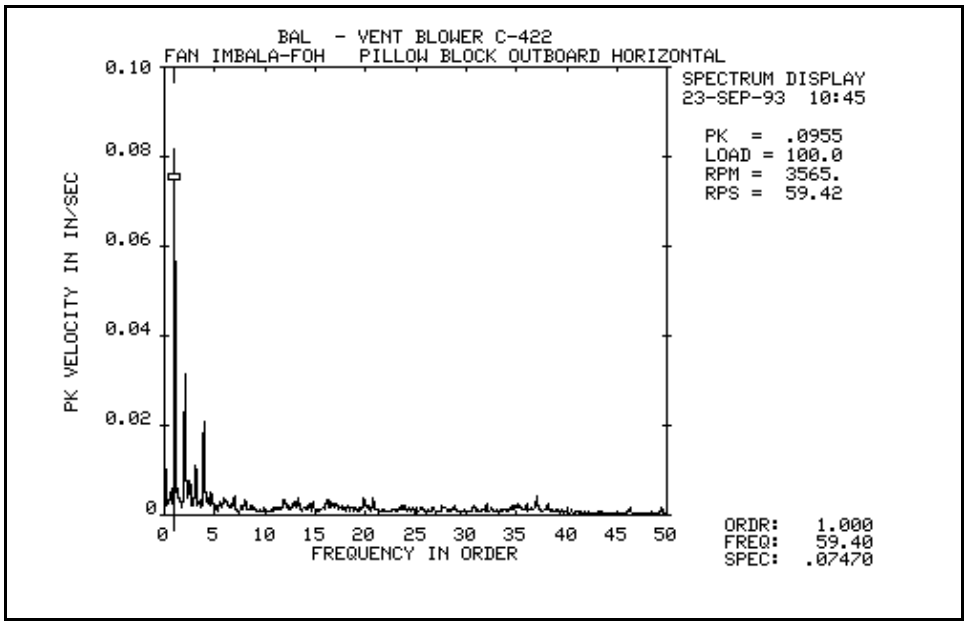
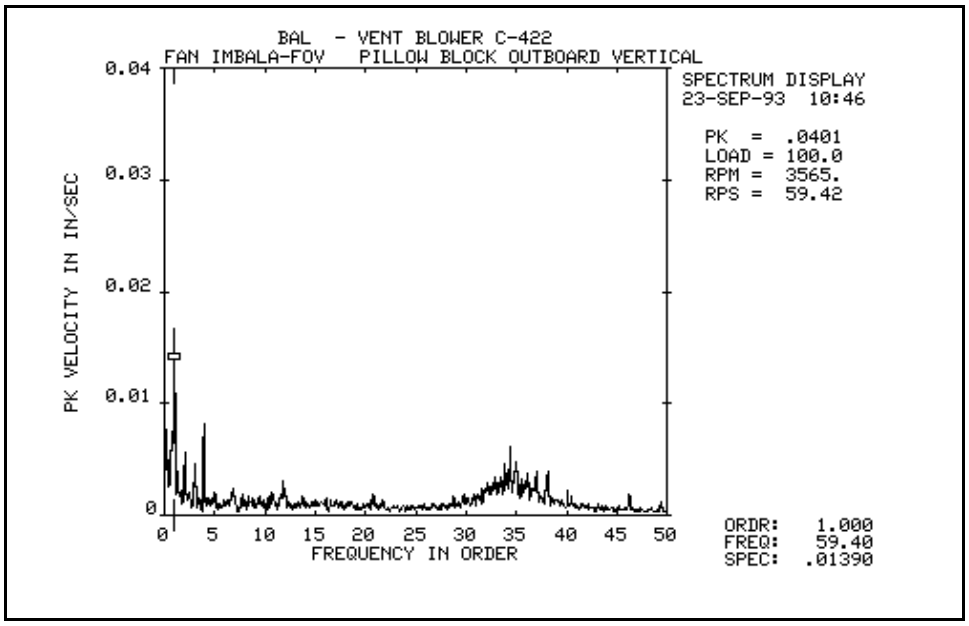


Notice the decrease in the amplitude of the 1xTS peak and the looseness on both measurement points. This clearly shows what happens to vibration when its driving force is removed. The unbalance appears to have damaged the bearing.



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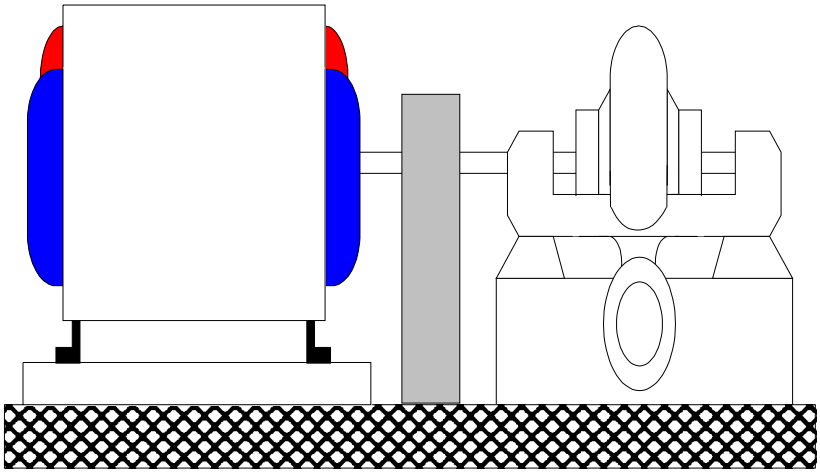
Unbalance Case #2 - West Desuperheater Pump



❶ 75-HP, 3600-RPM motor.

❷ Unit had frequent pump failures.

❸ Motor had very few hours of run time.

❹ Vibration analysis was used as a last resort to determine the problem.

❺ Data was collected with the motor uncoupled.

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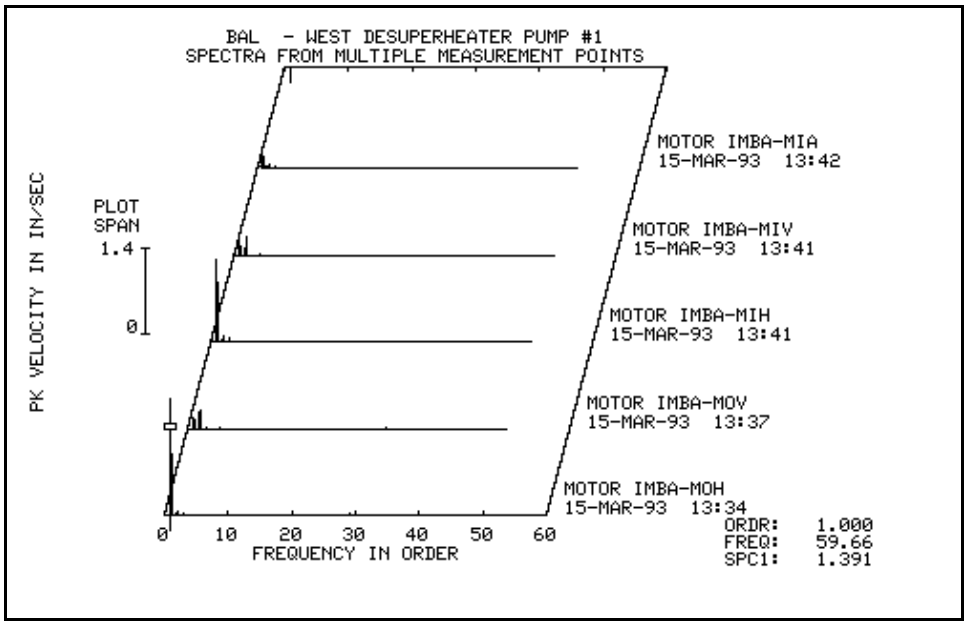
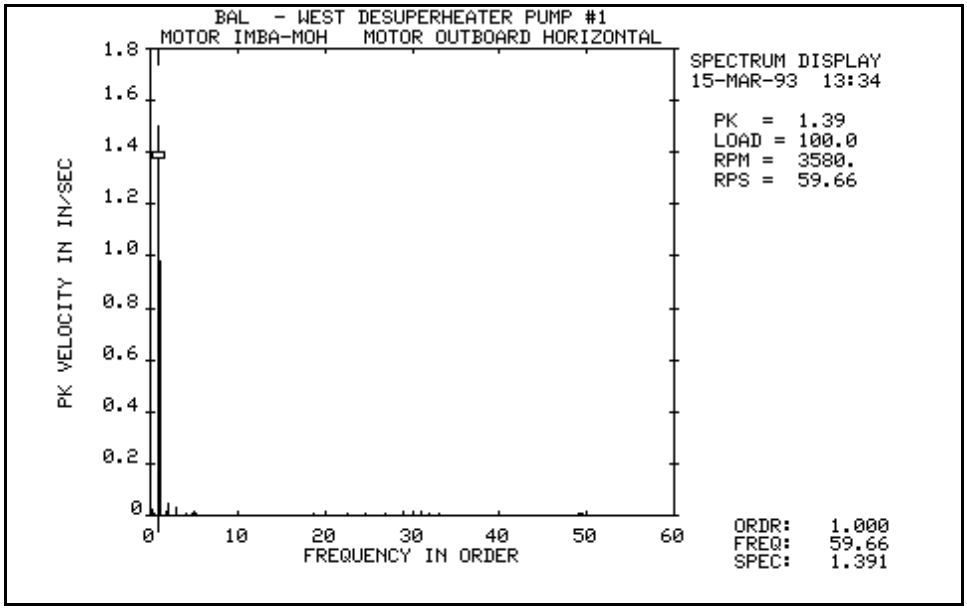


The motor points are displayed on the multiple plot above. The running speed peak is marked and shows an amplitude of 1.3 in/sec (IPS). It appears that the highest amplitudes are in the horizontal direction. Remember, the motor is running uncoupled from the pump.



Look at the amplitude of the 1xTS peak on the Motor Outboard Horizontal (MOH) point. As discussed, there are no harmonics of motor turning speed. What should the waveform look like?

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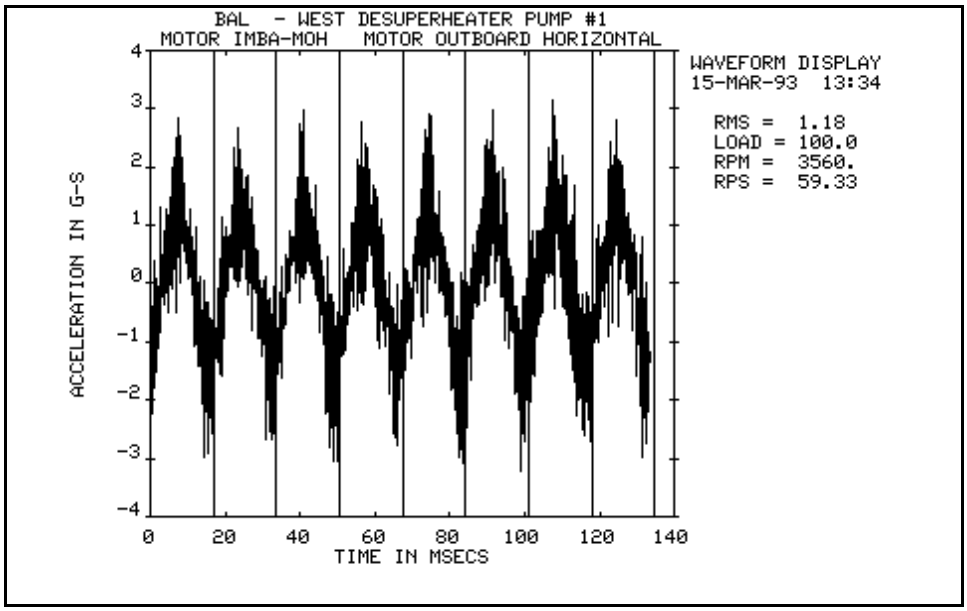
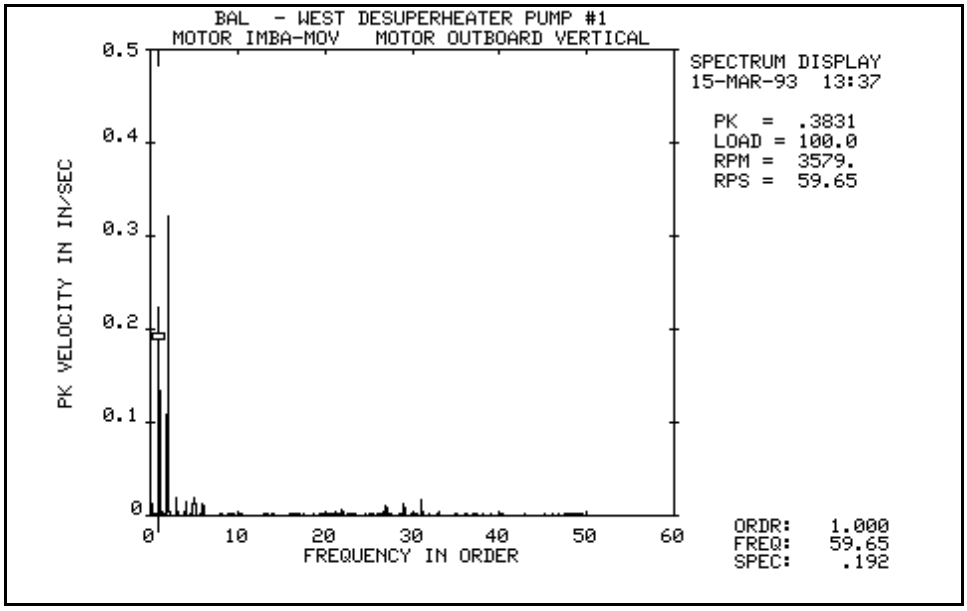


As expected, the waveform is repetitive, sinusoidal and has a strong 1xTS pattern. Notice the amplitude is fairly high, but lacks any impacting.



The spectrum from the Motor Outboard Vertical (MOV) measurement point is shown above. Its amplitude is 6.5 times less than that of the horizontal reading. This is a good indication that unbalance is not the problem that needs to be corrected. When the motor was removed for balancing, it was found the foundation grouting was cracked. Anytime you find a 3:1 ratio when comparing the horizontal to vertical reading, suspect another problem other than unbalance.

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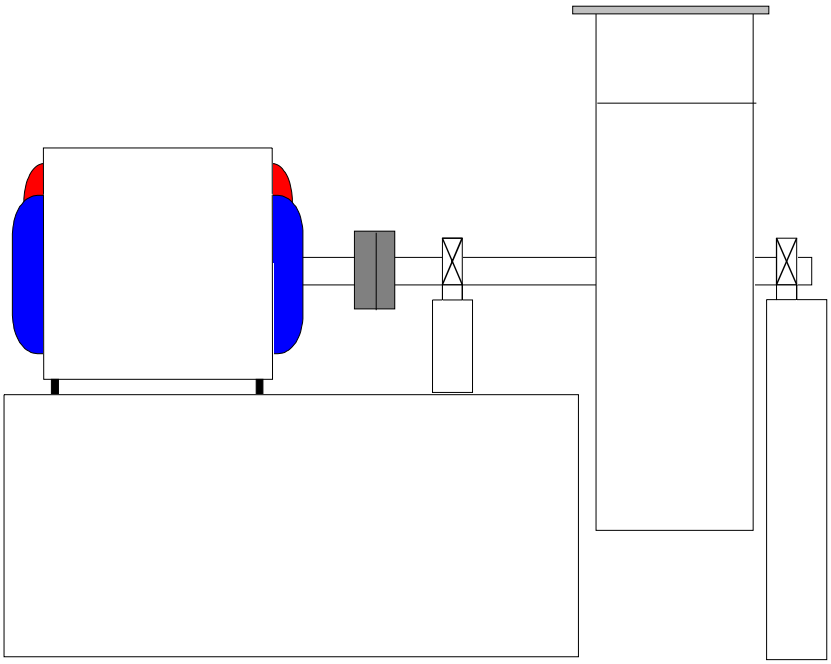
Unbalance Case #3 - Reactor Fan #6



❶ 250-HP, 1200-RPM motor.

❷ Fan is center-hung.

❸ Highest 1xTS vibration on horizontal readings.

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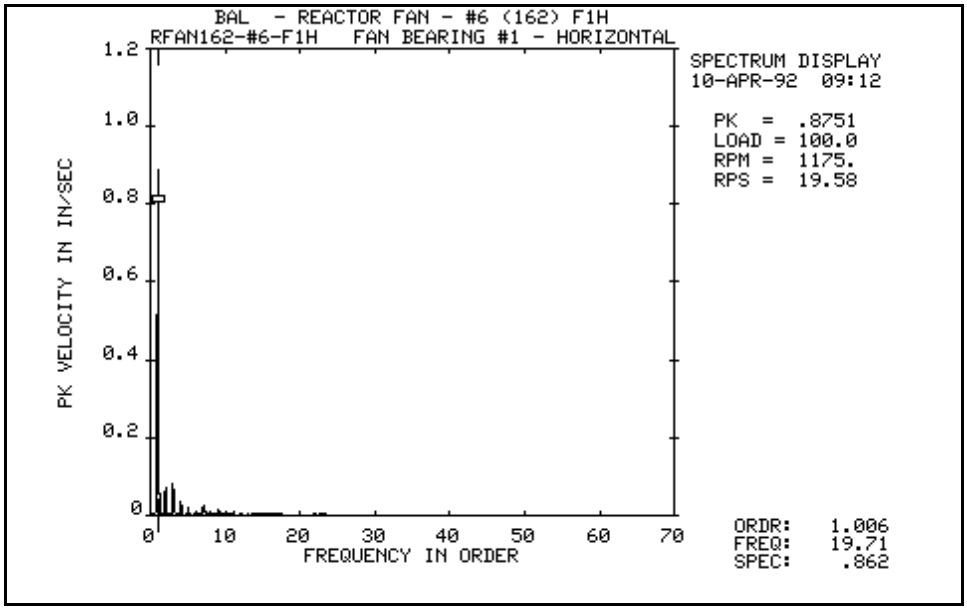
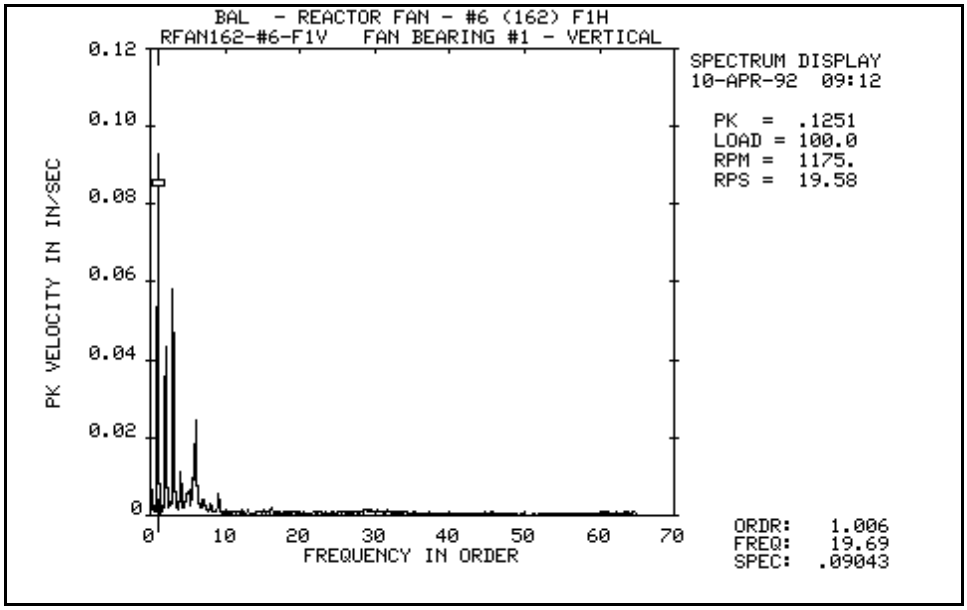


The Fan Inboard Horizontal (FIH) measurement point is shown above. Once again, the 1xTS peak is the dominant peak in the spectrum with little or no harmonics. The Fan Inboard Vertical (FIV) spectrum is displayed.



This plot also shows the 1xTS peak as the dominant peak in the spectrum. The amplitude of the FIH 1xTS peak is almost 10 times higher than the FIV 1xTS peak. This should initiate the collection of more data to assist in making the proper diagnosis. The next page shows a measurement point that should be considered.

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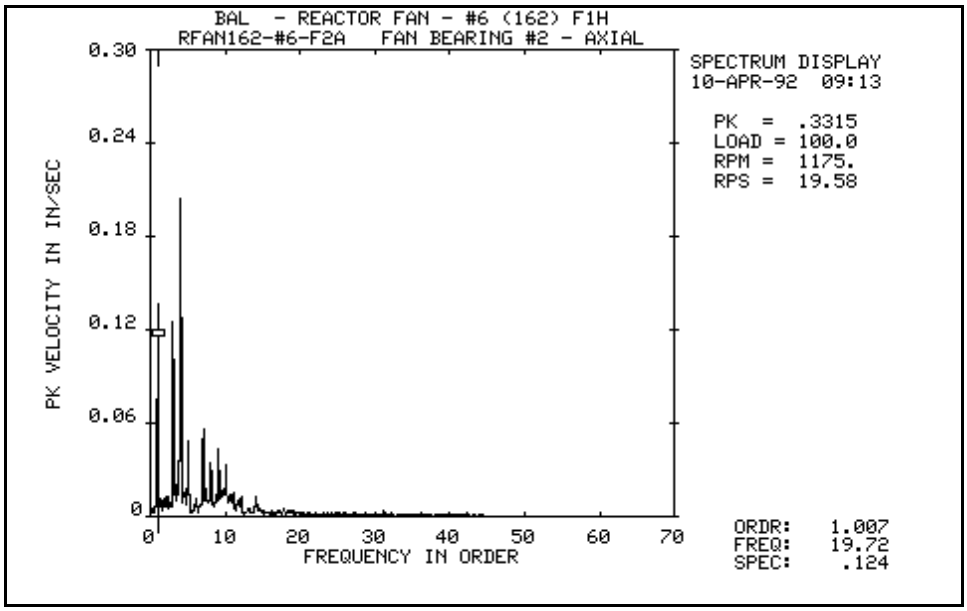

The fan axial measurement point is displayed above. Notice the 1xTS peak is higher than the vertical 1xTS peak, however, it is still much lower than the horizontal 1xTS peak. Further investigation should continue to determine the cause of the high ratios horizontal to vertical.

Could this problem have been misalignment?

Without the benefit of phase data or the time waveform this example is more difficult to make a call on.

Use all the tools that are available to you as an Analyst. Store the time waveforms and collect phase data to verify any suspected problem.

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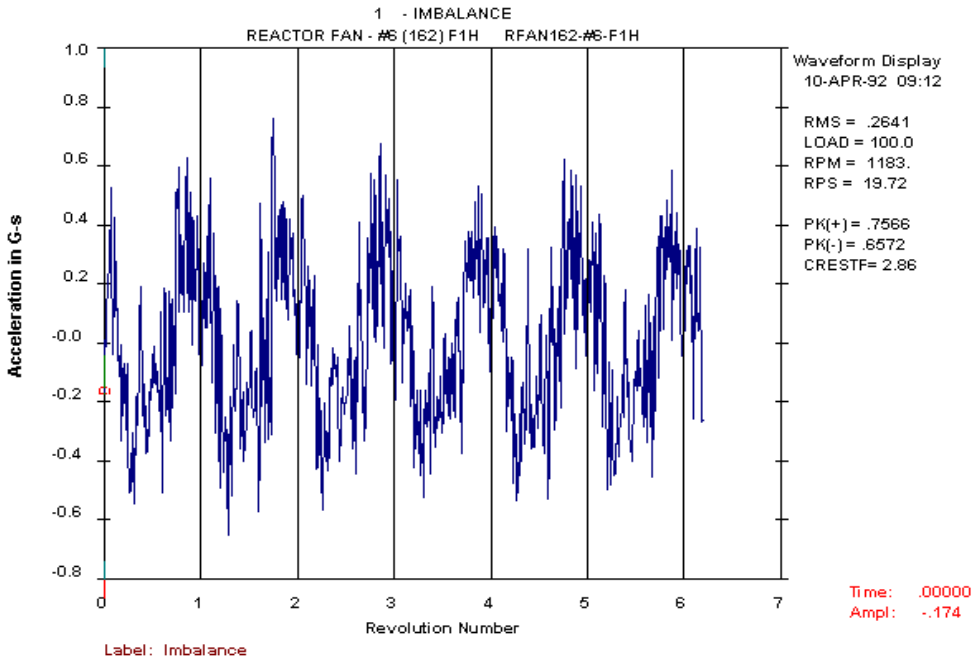
The waveform does help us to make a call on this example. We can be more confident now that the problem was due to unbalance.



Again phase data would have helped us verify this problem.

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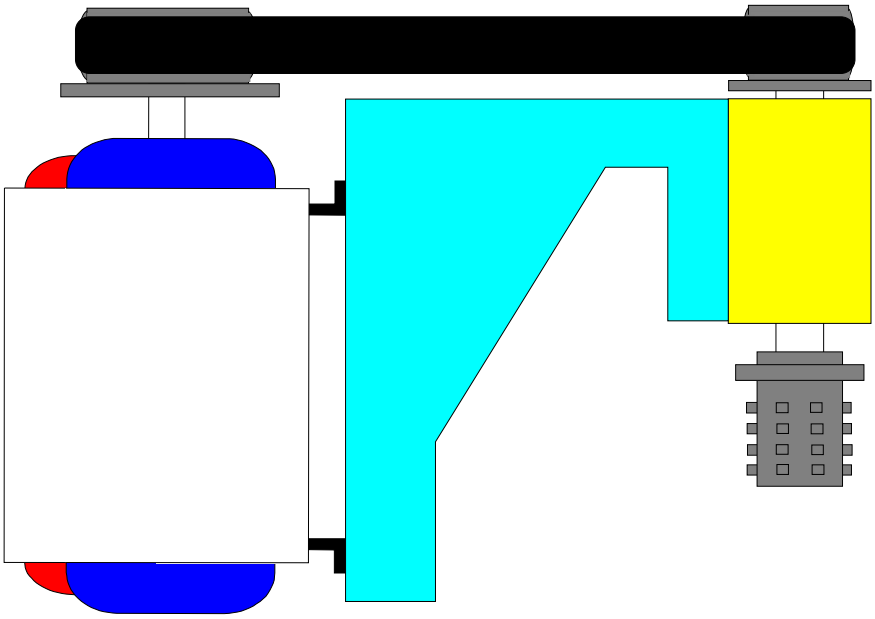
Unbalance Case #4 - Office Chair Shaper



❶ 50-HP, 3600-RPM motor.

❷ Unit is driven by a flat belt.

❸ The Sheave Ratio is 2:1.

❹ The shaper had excessive vibration.

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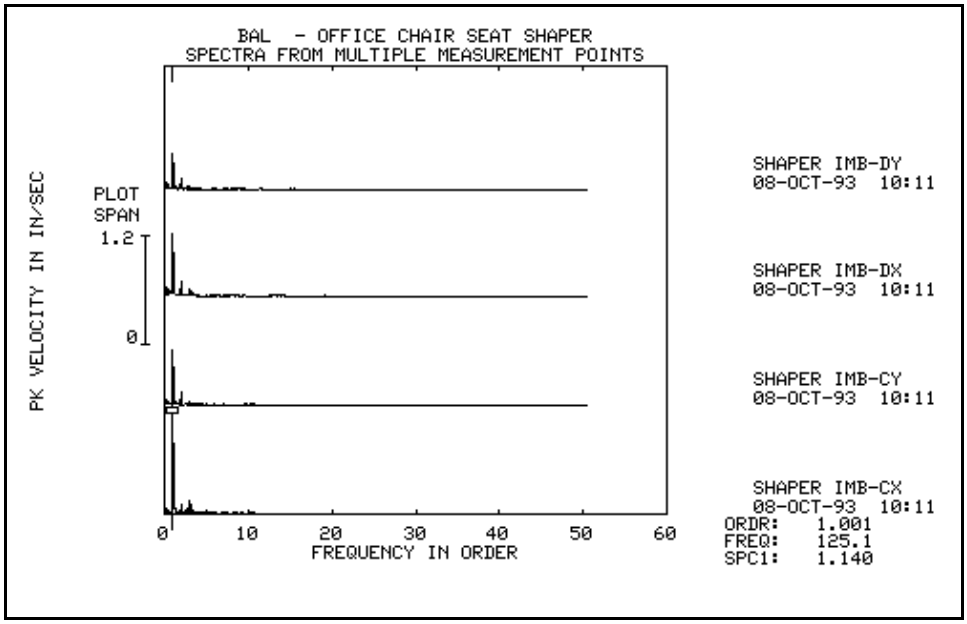
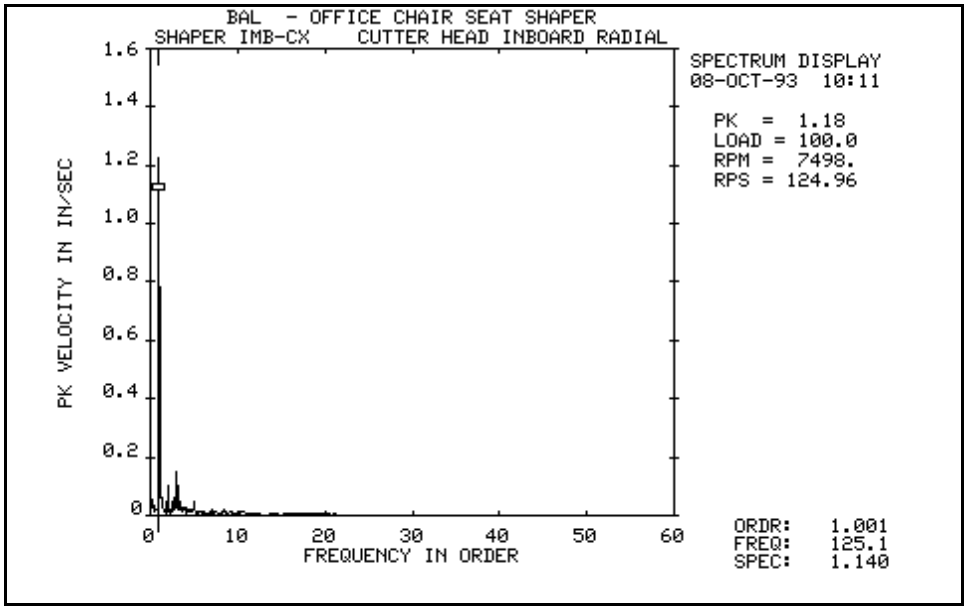


The multiple-point plot is displayed above for the shaper measurement locations. Measurement point CX has the highest amplitude reading over 1.0 IPS at shaper running speed.



The spectrum for measurement point CX is displayed above. The characteristics for unbalance are again seen in this plot, which is a dominant 1xTS peak with little or no turning speed harmonics.

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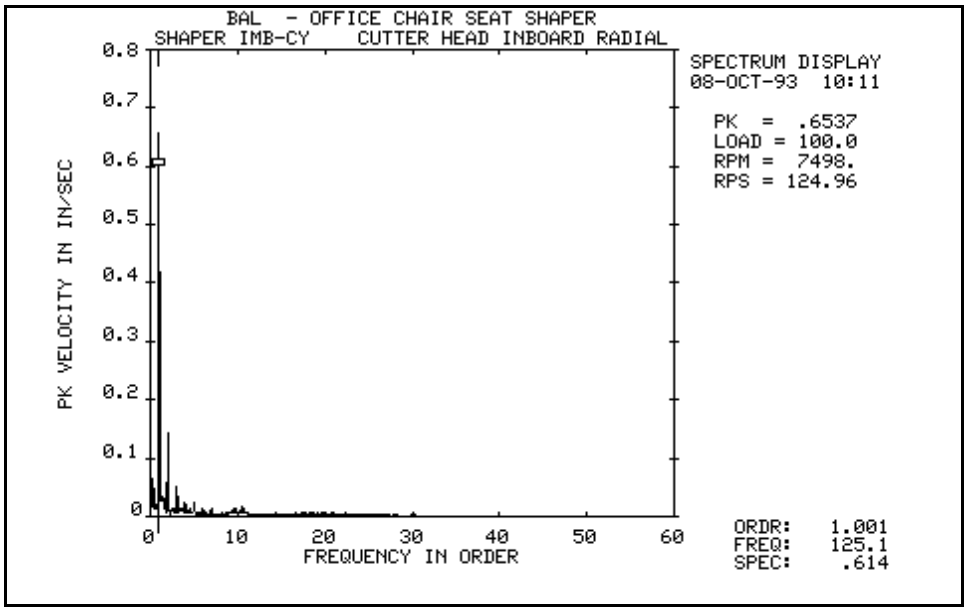
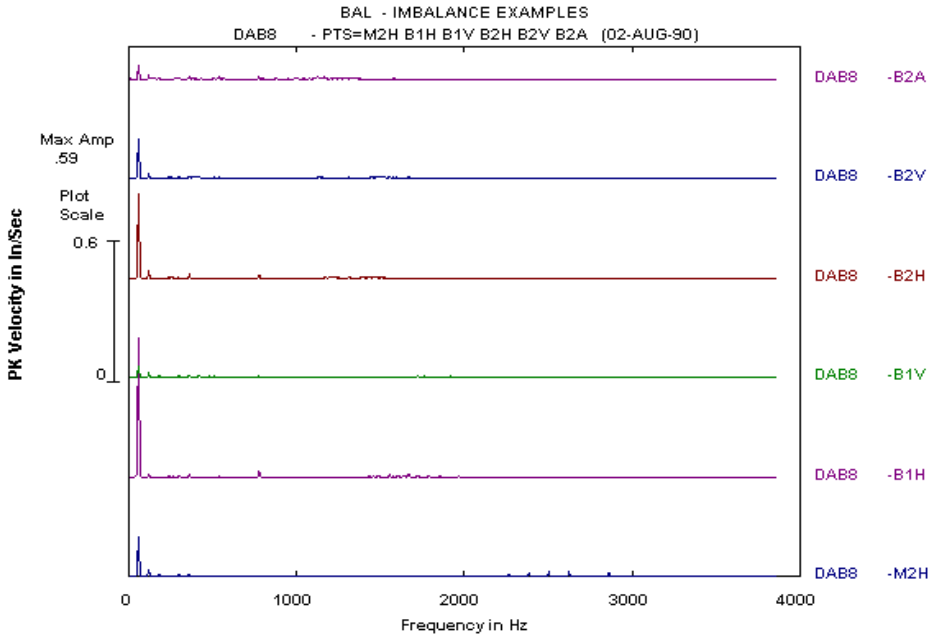


The spectrum for measurement point CY is displayed above. Its amplitude is consistent with what would be expected when it is compared to the horizontal reading. Remember, belt-driven systems can have high ratios when the horizontal and vertical amplitudes are compared.



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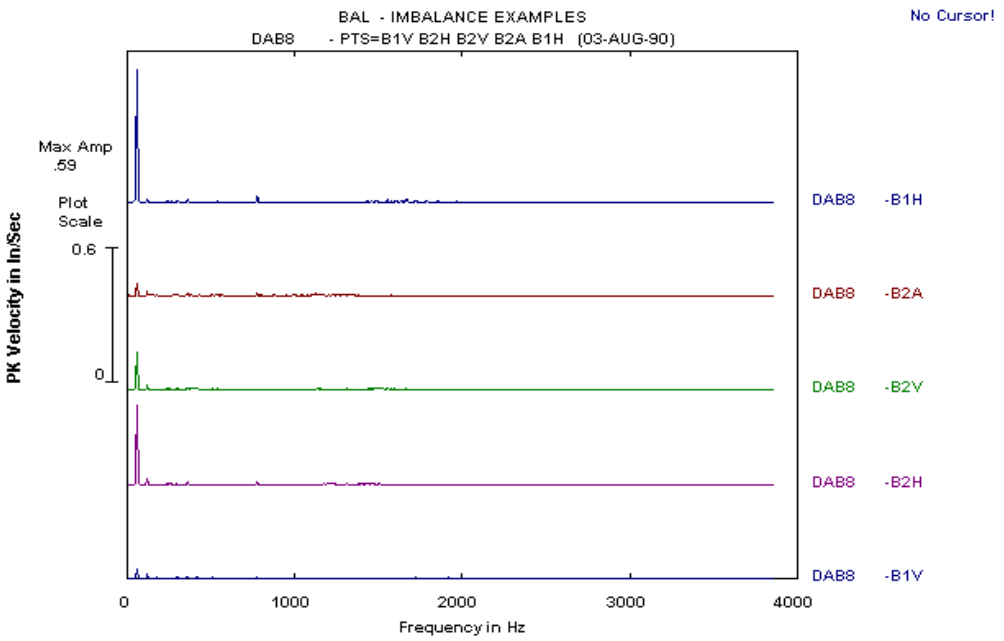
Unbalance Case #5 - Dunson Blower

Displayed below is a multiple points plot of the Blower. Where is most of the vibration? In what direction? This plot also includes the motor inboard measurement point.



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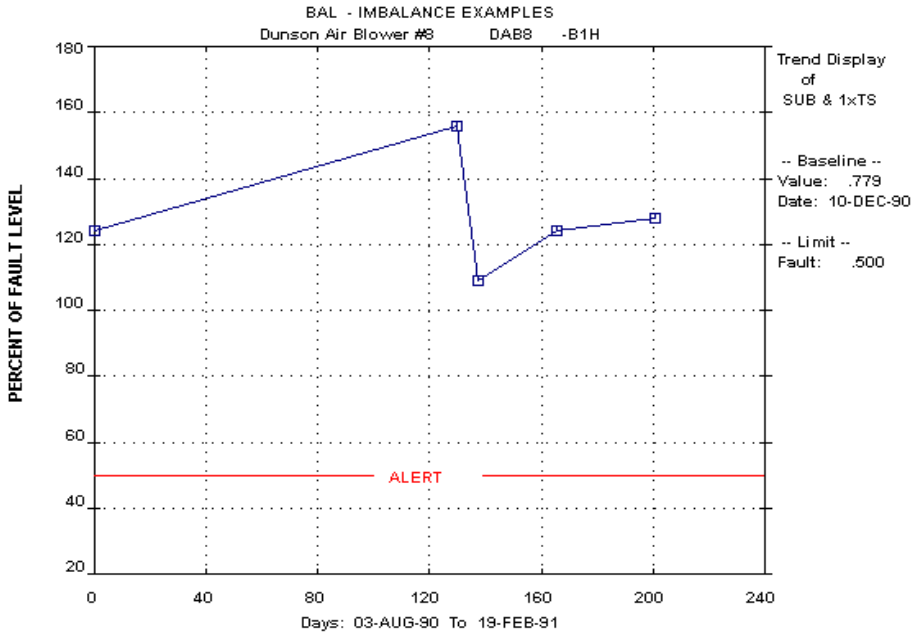
Displayed below is a parameter profile trend of the BIH measurement point of just the 1 x rpm band.



Note that the blower had been running well above the alert level that was defined for this unit.

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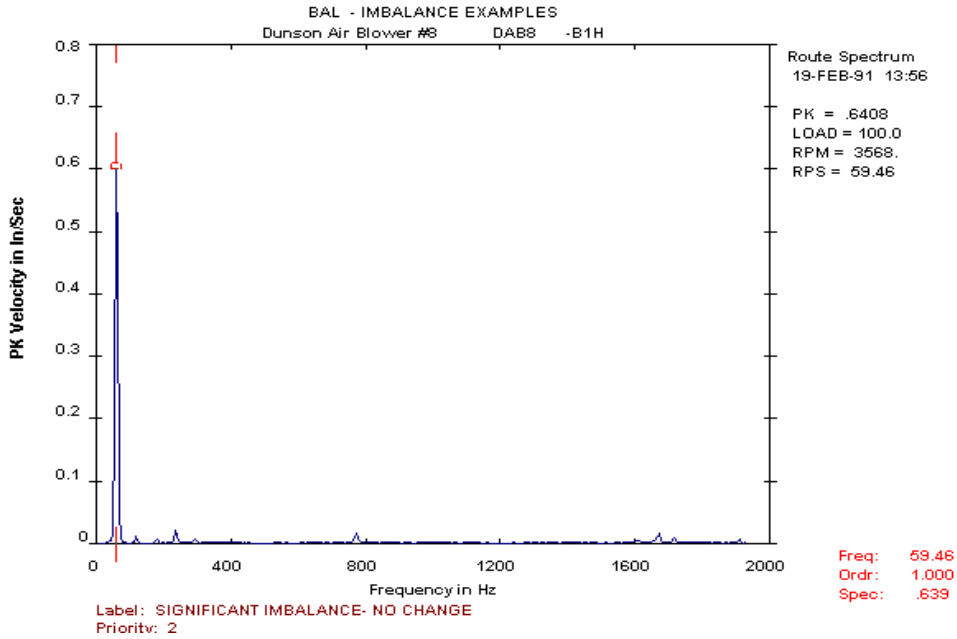
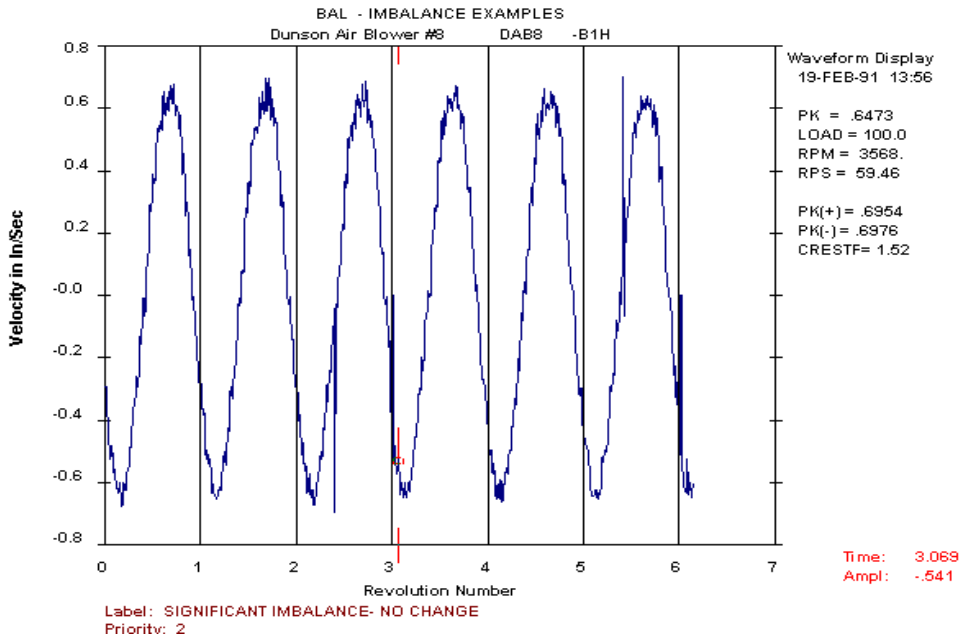
This single full scale spectrum also reveals almost all of the vibration is at 1 x rpm.



The waveform for this spectrum displays a sinusoidal 1 x rpm per revolution type pattern. Note that this is in the radial (horizontal direction). Armed with both the spectral and waveform data the analyst can feel confident in making the correct call on this machine. When the machine was taken down and balanced it ran at less than .1 in/sec.



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Section Five
Misalignment

Objectives

- Identify misalignment based on spectral and waveform characteristics.
- Explain how to use phase to help distinguish misalignment from unbalance.
- Describe analysis techniques using single-channel data acquisition.

What is Misalignment?

Misalignment is another major cause of machine component damage. At one time, it was considered the second most prevalent vibration source. Most industries with predictive maintenance programs, however, have found that misalignment occurs more often than unbalance.

Maintenance personnel too often rely on manufacturers' claims that their self-aligning bearings and flexible couplings compensate for shaft misalignment. Nevertheless, major vibration problems can be present even when couplings are aligned within manufacturers' tolerances. It is important for those responsible for machine installation and maintenance to understand both the importance of shaft alignment and the proper methods of aligning shafts.

Misalignment occurs because of poor alignment between mating pieces such as coupling halves, bearings, shafts and pulleys. Misalignment of couplings, bearings and pulleys results in high axial and radial vibration readings. The axial readings amplitude can be as much as 0.5 to 2 times the amplitude of the radial readings. The radial readings can appear at 1x, 2x, 3x, and even higher multiples of shaft turning speed. This is especially true when a coupling defect is present. On certain types of equipment, however, the predominant vibration occurs at 1xTS and can be confused with unbalance. Phase data is a valuable tool in distinguishing between an unbalance or a misalignment problem.

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Characteristics of Misalignment

- High axial levels
- High radial vibration at 1x, 2x, 3x, 4x, or higher multiples of turning speed
- 180° phase difference in axial or radial directions across the coupling
- Repeatable periodic time waveform with 1, 2, 3, or 4 clear peaks per revolution
- Excessive bearing temperatures
- Premature bearing failures
- Coupling failures
- Oil leakage at the seal
- Excessive amounts of grease on the coupling guard
- Shaft breakage at or close to the inboard bearings

Shaft Center Line Misalignment

Two types of shaft center line misalignment are discussed below — *angular* and *offset* or *parallel* misalignment. Most misalignment problems are a combination of both.

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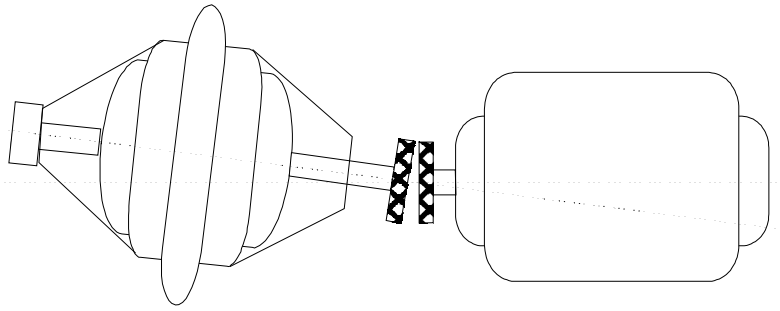
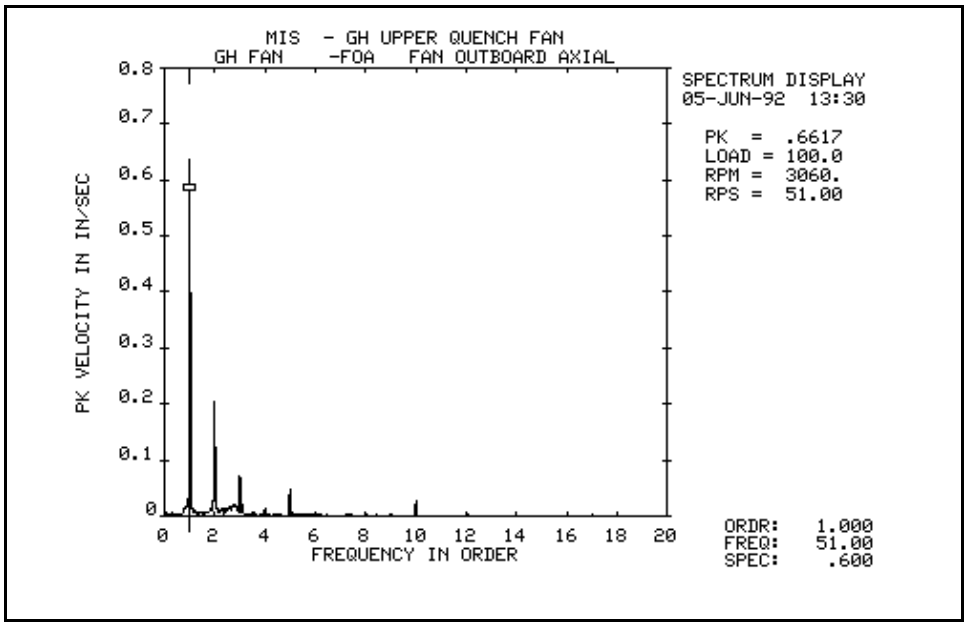
Angular Misalignment



Angular misalignment occurs when the center lines of two shafts meet at an angle. The shafts can be angled in the vertical or horizontal planes. The spectrum typically has a dominant peak at 1xTS in the axial direction. Radial readings in the spectrum will show peaks at 1xTS and 2xTS if offset misalignment is also present. The highest vibration usually occurs in the direction of the angularity, which means that if the angularity occurs in the horizontal direction, the highest vibration should also appear in the horizontal direction.



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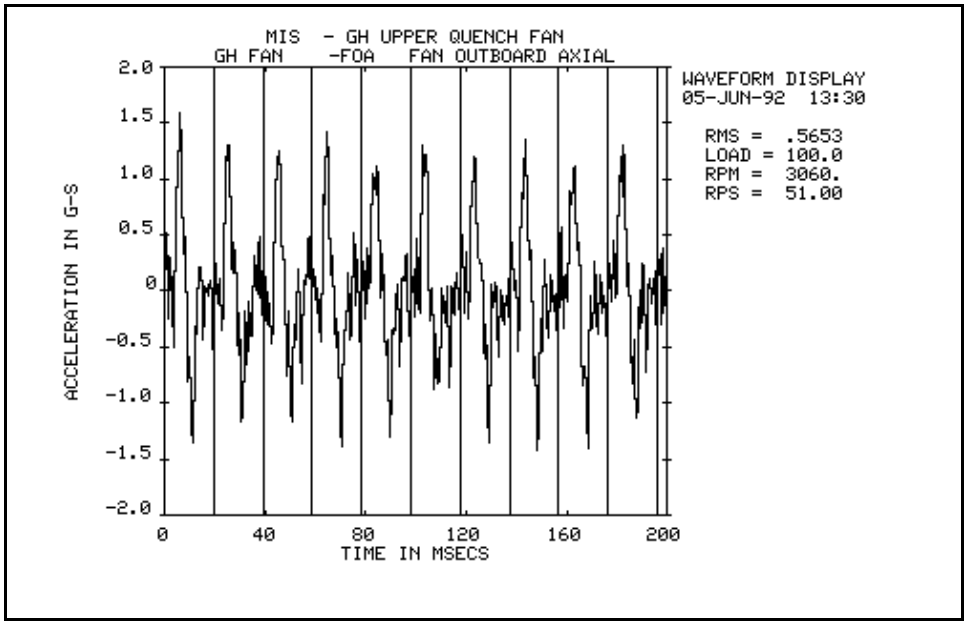
The waveform characteristically is sinusoidal with one or two clear cycles per revolution. The waveform amplitude in g's is low and does not show much impacting.



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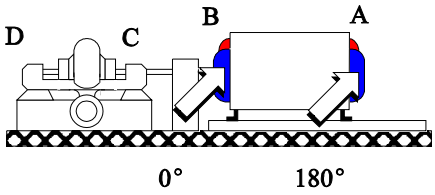
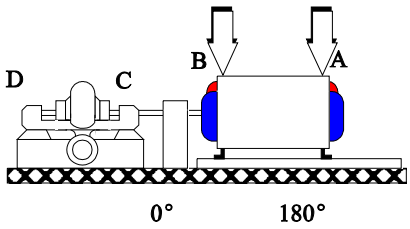
The typical **phase** relationships for angular misalignment are:

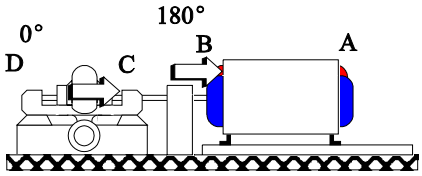
- The ***horizontal phase relationship*** from A to B is 180° ± 30°.



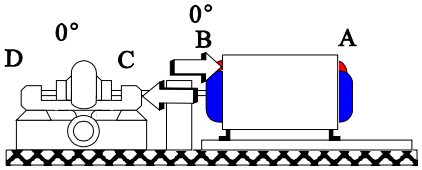
- The ***vertical phase relationship*** from A to B is 180° ± 30°.



- The ***axial phase relationship*** from B to C is 180° ± 30°.



Note: Do not forget to take the transducer orientation into consideration on axial readings. Confusion here could lead to making the wrong call.



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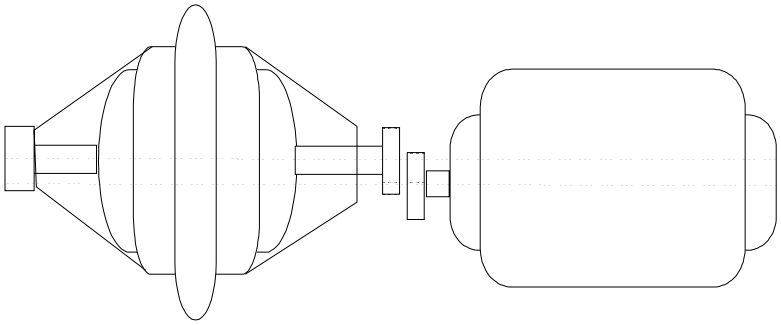
Offset Misalignment



With ***offset misalignment***, the shafts are parallel but displaced from each other. The shafts can be offset in the vertical or horizontal planes. The spectrum shows a predominant 2xTS peak in the radial direction. The highest vibration usually occurs in the direction of the offset — if the offset occurs in the vertical direction, the highest vibration should also appear in the vertical direction. Since the coupling forces the two offset shafts to meet, it introduces some angular misalignment. This can cause a 1xTS peak in the radial and axial direction that exceeds the 2xTS peak.

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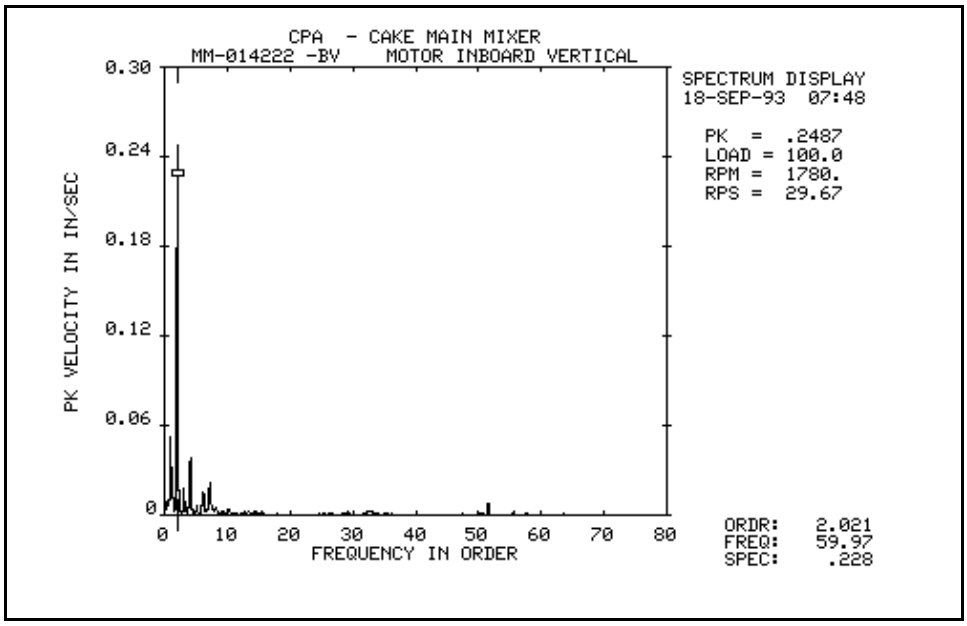
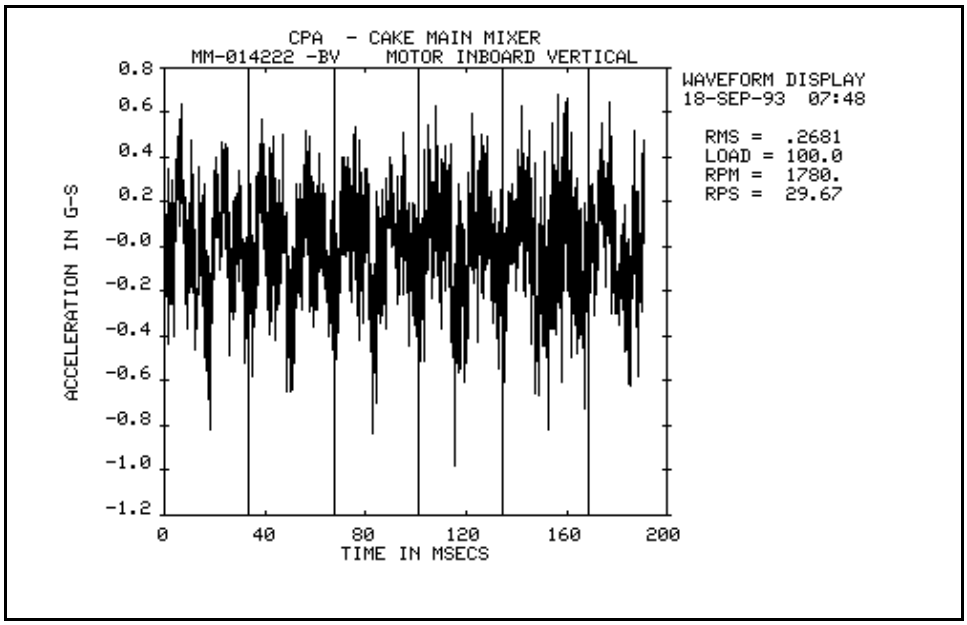


The waveform is characteristically sinusoidal with one or two clear cycles per revolution. The waveform amplitude in g's is low and does not show much impacting.



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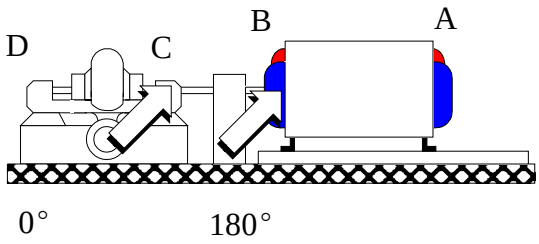
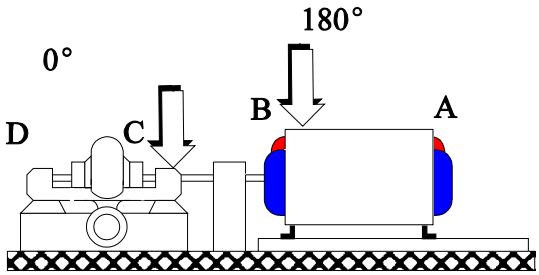
The typical **phase** relationships for Offset Misalignment are:

- The horizontal phase relationship from B to C is equal to 180° ± 30°.



- The vertical phase relationship from B to C is equal to 180° ± 30°.



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Shaft Component Misalignment

A partial list of possibly misaligned components are listed below:

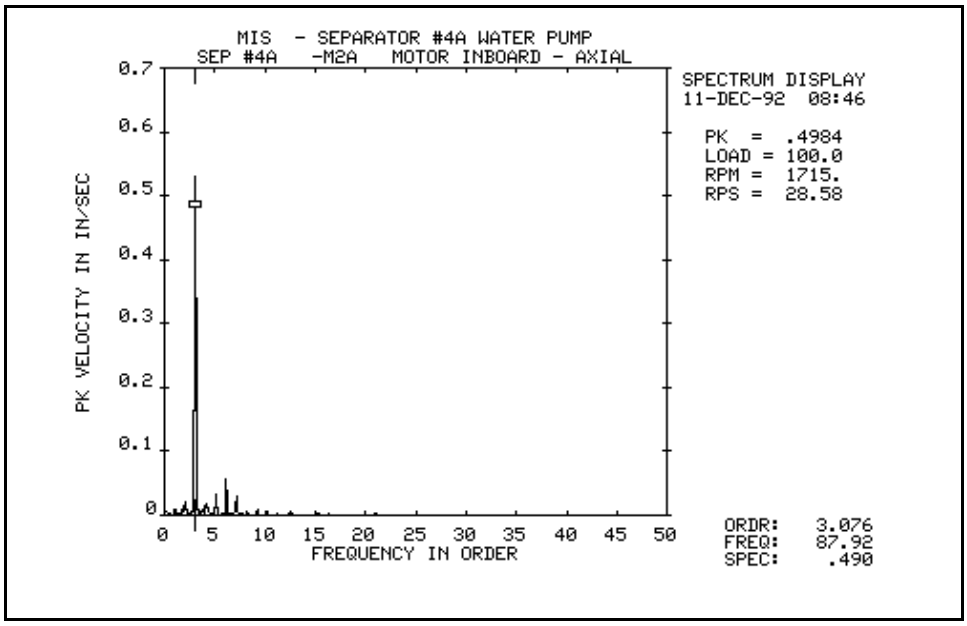
❶ Pulleys
❷ Gears
❸ Couplings
❹ Bearings

This section will cover only couplings and bearings. Pulleys and gears are discussed in detail later in this course.

Coupling Misalignment

Coupling misalignment occurs when two shafts are misaligned, the coupling is worn, or the coupling was installed improperly. The coupling can be misaligned vertically or horizontally. The spectrum can have multiples of 1x to 6xTS and harmonics in both the axial and radial directions depending on the type of coupling used. Once again, the highest vibration usually occurs in the direction of the defect.

The appearance of the waveform will be dependent on the fault type and the coupling type. It can vary from very periodic and sinusoidal to very random with impacting.



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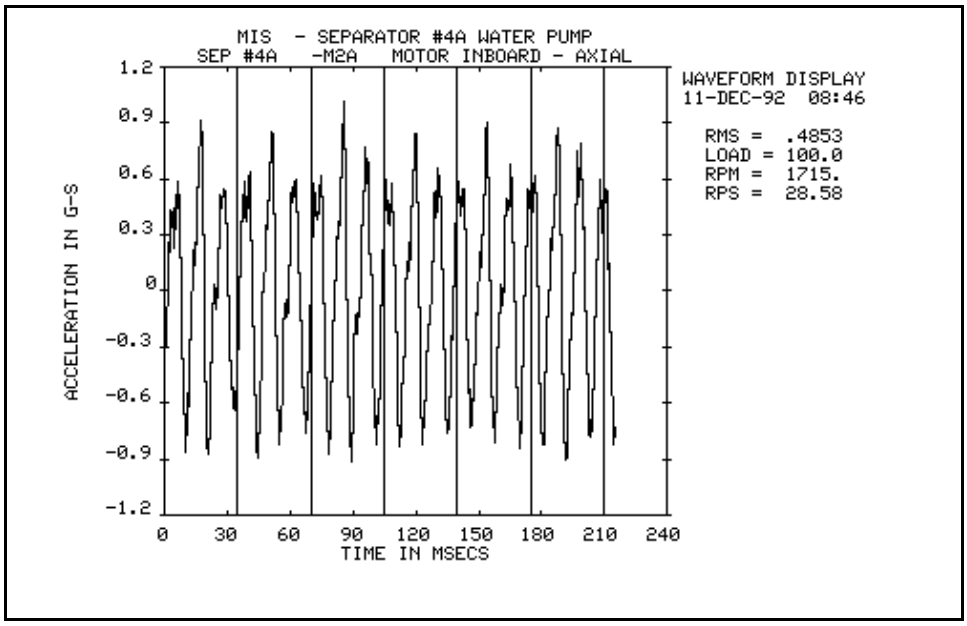
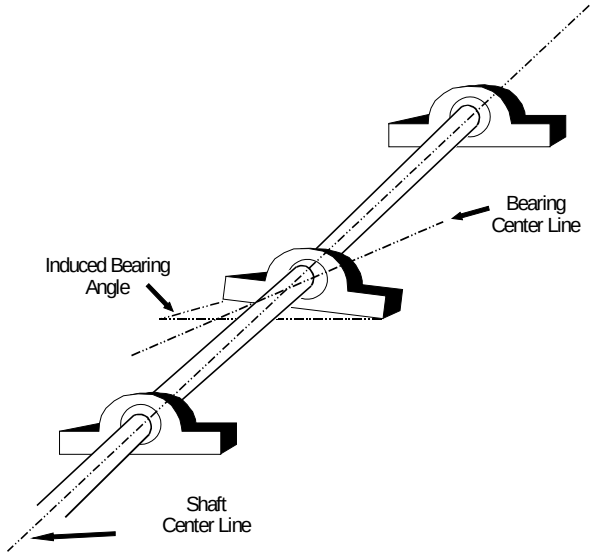


Phase readings will typically be a combination of the offset and angular characteristics.

Bearing Misalignment



Bearing misalignment occurs when the bearings are not mounted in the same plane; they are cocked relative to the shaft or if the machine distorts due to thermal growth or soft foot. The spectrum normally has the highest peak at 1xTS, 2xTS, or the number of rollers xTS in the axial direction.

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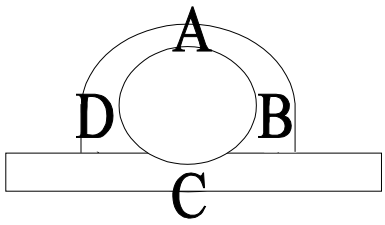
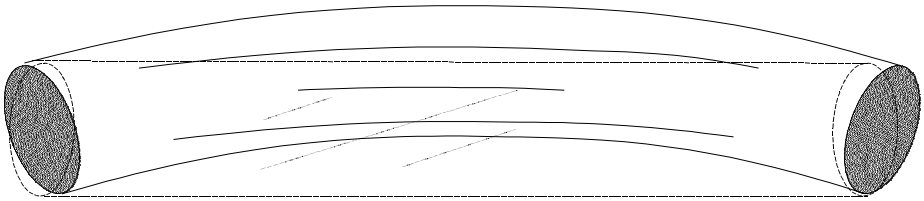
The waveform often shows a truncated or flattened pattern indicating a rub. It may also appear periodic and sinusoidal with low amplitude.

The typical phase relationships of bearing misalignment are:

- The axial phase relationship from A to C is equal to 180°± 20°.
- The axial phase relationship from B to D is equal to 180° ± 20°.



Bent Shaft



Vibration can vary depending on where the shaft is bent. If the shaft is bent near the center, the spectrum has a dominant peak at 1xTS in the radial and axial direction. It can, however, appear at 2xTS if the shaft is bent near the coupling. When the rotor is spun up past the first critical, the amplitude will continue to rise exponentially. When speed is reduced, the amplitude will decrease drastically. If this was due to imbalance, the amplitude would not decrease as fast. Remember, this is above the first critical.

The waveform will resemble a mixture of misalignment and imbalance. Keep in mind that the shaft is actually out of balance.

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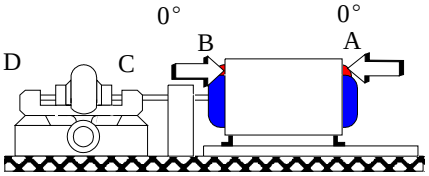
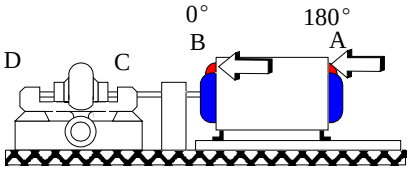
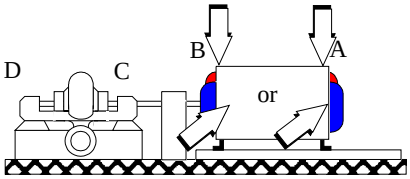
The phase characteristics of a bent shaft are:

- The axial phase relationship from A to B is equal to 180° ± 30°.



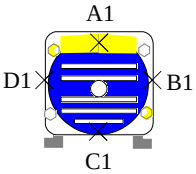


- The horizontal-to-vertical phase relationship on A and B is equal to 90° or 270° ± 30°.



90° or 270°
Horizontal to Vertical

- The axial end phase relationship from A1 to C1 and B1 to D1 is equal to 180° ± 30°.



Axial End Phase

Remember that the phase data will be 180° out across the bend, not across the coupling.

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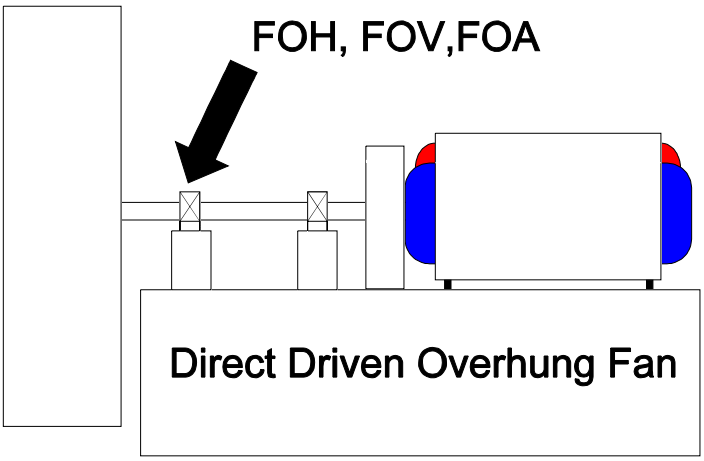
Case Histories - Misalignment

Misalignment Case #1 - Quench Fan



❶ 100-HP, variable-speed Motor.

❷ Direct-driven, overhung fan.

❸ High axial readings on Fan Outboard bearing point.

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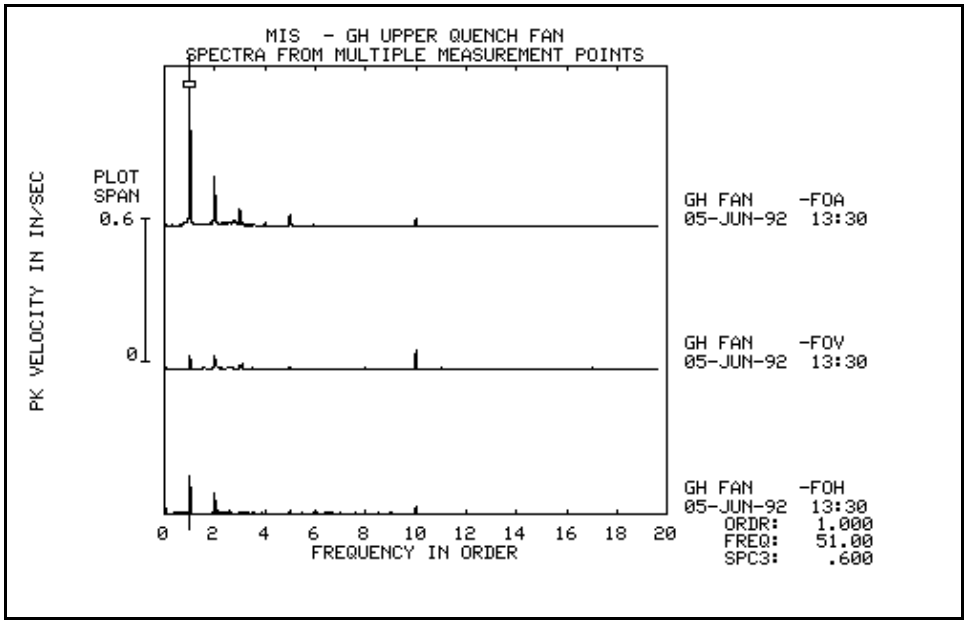
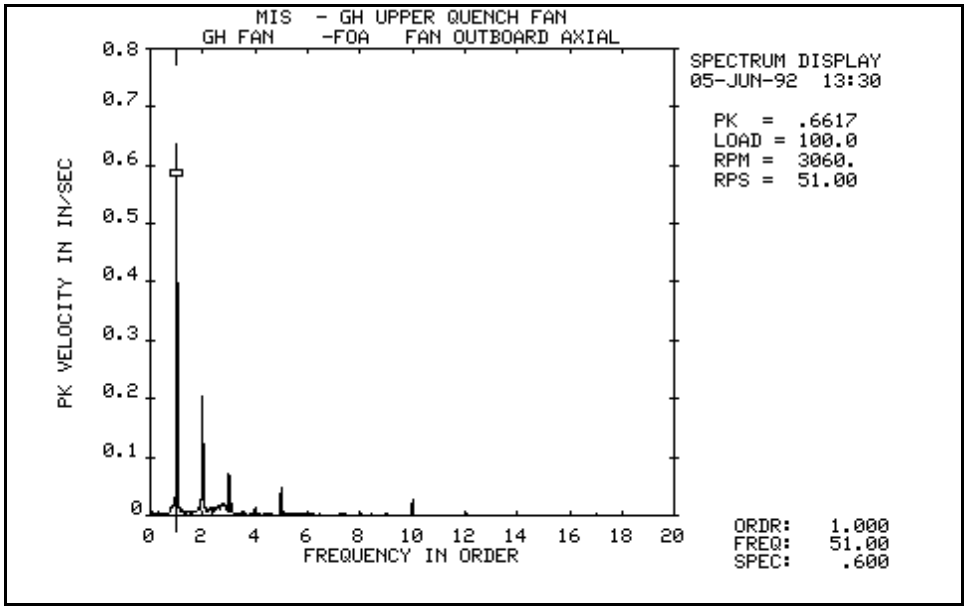


The multiple plot above came from a case history involving a fan. The data displayed is from the outboard fan bearing. Notice that the highest amplitude is in the axial direction at fan running speed. The spectrum for the fan axial point appears below.



A full-screen plot of the fan axial data is shown above. The cursor marks the running speed peak of the fan. Note that the axial vibration amplitude is six times the amplitude of the radial readings. If the axial measurement point was not taken it would have been difficult to diagnose this problem as angular misalignment.

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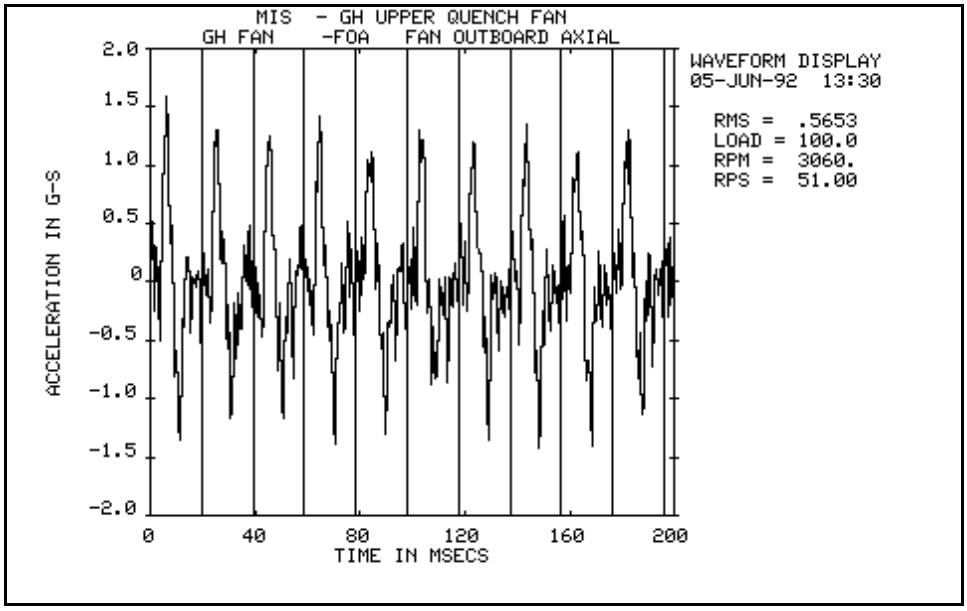

The time waveform for the fan outboard axial data helps to confirm the problem on this machine. Vertical lines on the plot show the time required for the fan shaft to complete one revolution. Evidence that the vibration is being driven by a repetitive force is demonstrated by the presence of one or two peaks per shaft revolution. As discussed, the waveform is repetitive and lacks complexity.

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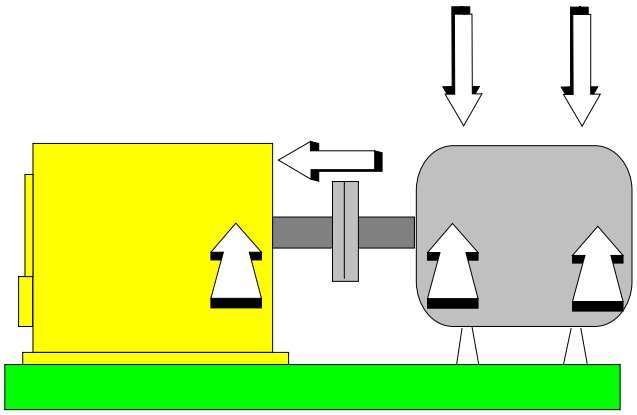
Misalignment Case #2 - Main Cake Mixer



❶ 75-HP, four-pole motor.

❷ The motor drives a double reduction gearbox.

❸ High vibration at motor running speed was present on all measurement points.

❹ The 1xTS vibration was primarily in the horizontal direction.

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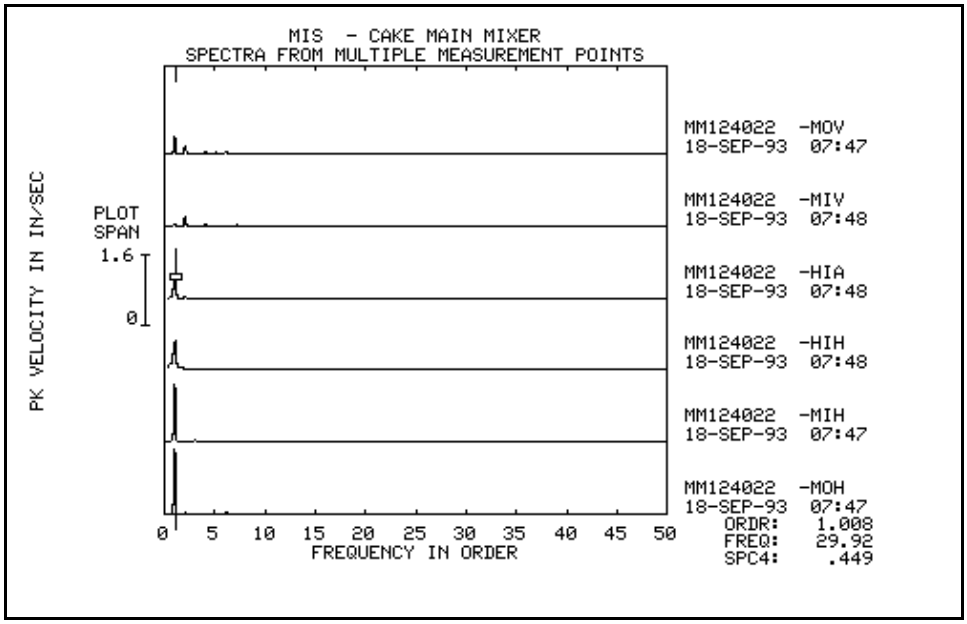
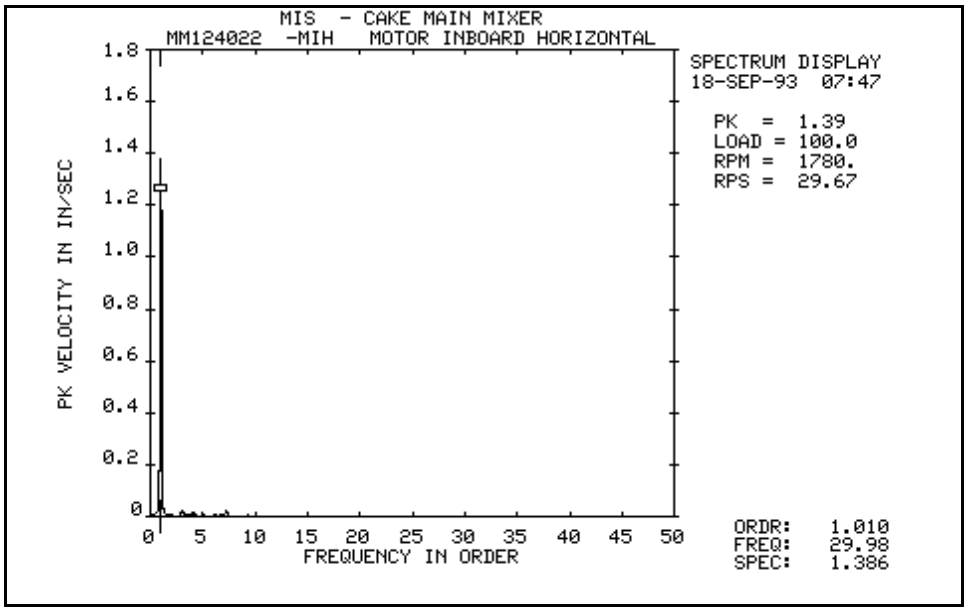


High 1xTS vibration amplitudes exist on the motor and gearbox in this example. The gearbox is direct-driven from the motor. Notice that the 1xTS vibration appears primarily in the horizontal direction with amplitudes exceeding 1.0 IPS.



The full-screen plot of the motor inboard horizontal spectrum is shown above. Severity of the defect is easy to determine based on the 1xTS amplitude exceeding 1.3 IPS.

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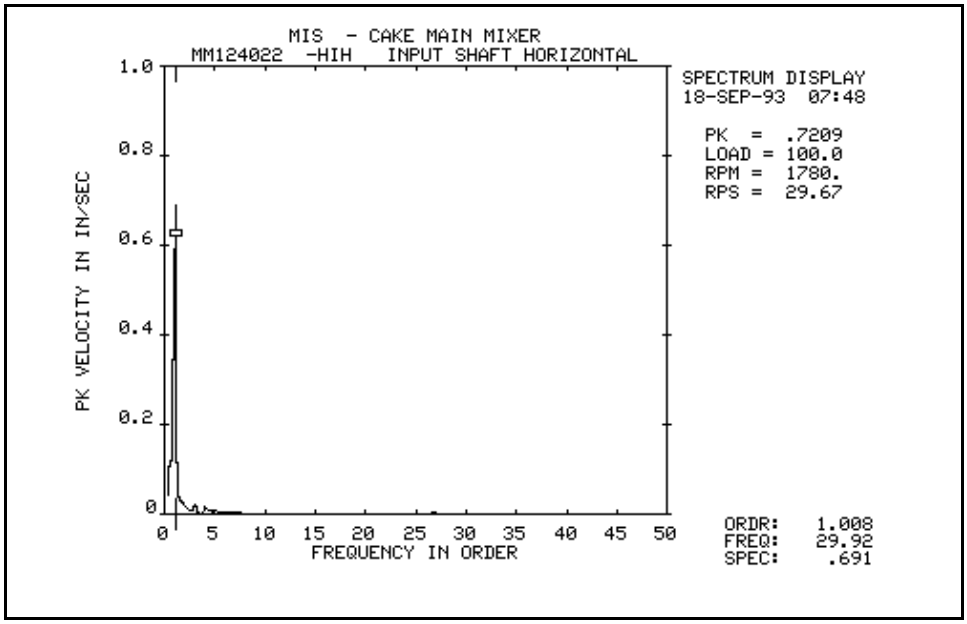

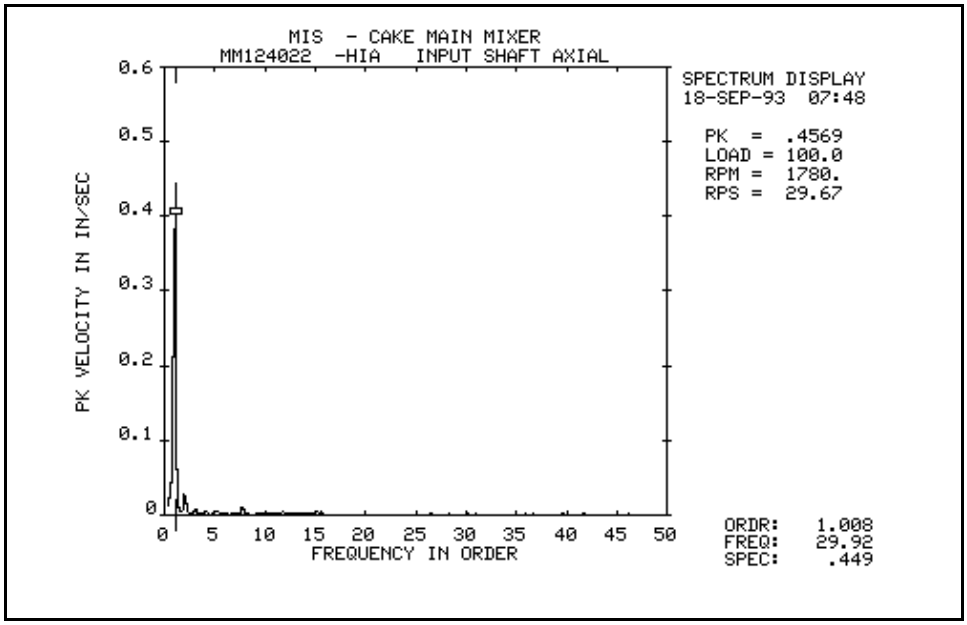

A horizontal and an axial reading from the gearbox input shaft are displayed. Again, the axial reading confirms that the defect is angular misalignment. In what direction is the unit angularly misaligned? The amplitude in the axial direction does not have to “exceed” amplitudes in the radial directions for angular misalignment. Just compare radial to axial ratio, if the axial amplitude is 50 percent of the radial direction, possible angular misalignment.

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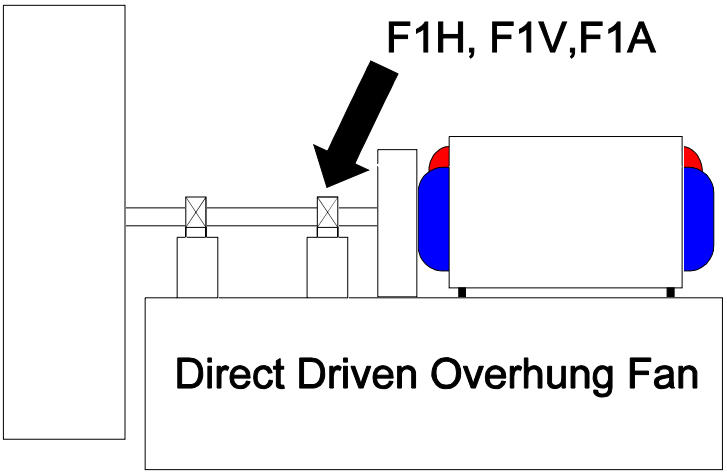
Misalignment Case #3 - Process Fan



❶ 250-HP Motor.

❷ High horizontal amplitude on fan bearing one.

❸ 0.025 to 0.030" runout was measured at the FOH point

❹ 1xTS in the horizontal direction.

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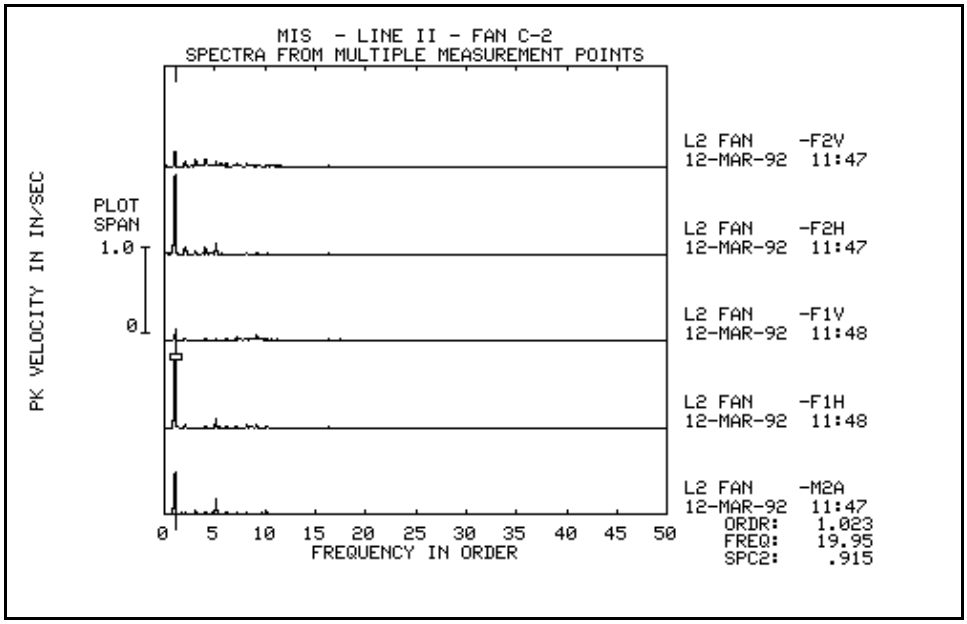
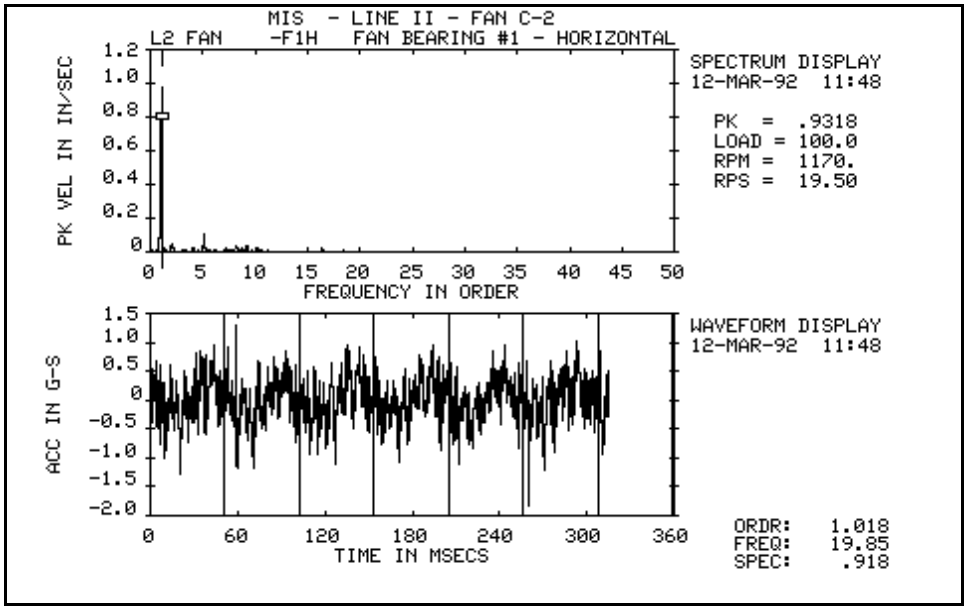


In the example above, peaks at running speed indicate a bent fan shaft. Fan Inboard Horizontal (F1H) appears to have the most energy. Note that the 1xTS vibration is primarily showing in the horizontal direction. Looks like unbalance. Phase data was a key to solving the problem. 180 degrees shift from top to bottom of bearing at FOH. 180 degrees side to side.



The spectrum and waveform above are from measurement point F1H. Notice the lack of any multiples of shaft turning speed in the spectrum. The waveform displays a repeatable, sinusoidal pattern.

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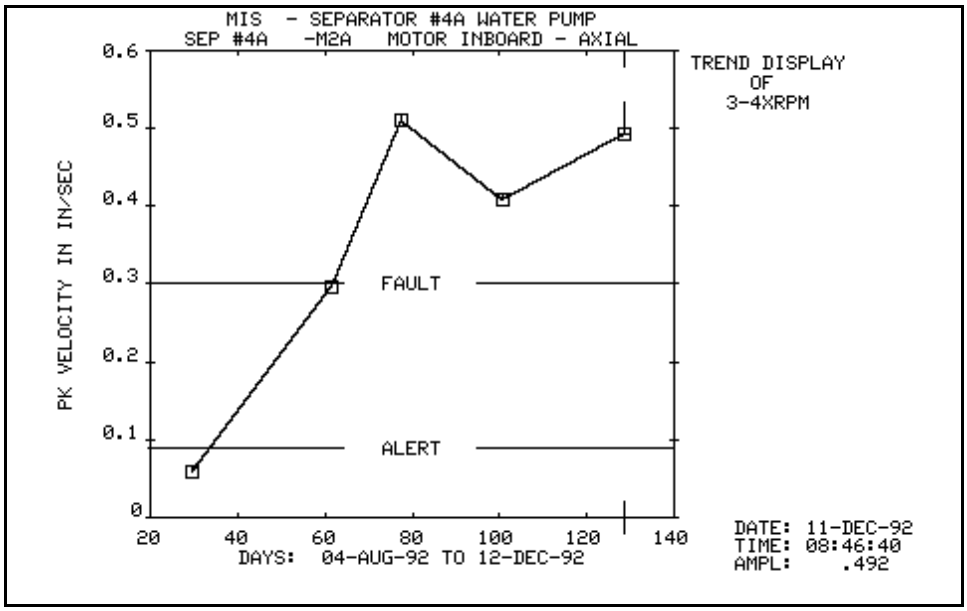
Misalignment Case #4 - Separator Water Pump #1

❶ 15-HP motor, 1800 RPM.

❷ Flexible type coupling. Direct Drive, Centerhung Pump

❸ High 3xTS vibration.



The trend plot for motor's 3 to 4xTS band is shown above. Look at how the amplitude has increased over the past four collection periods. This increase in vibration is from angular misalignment.

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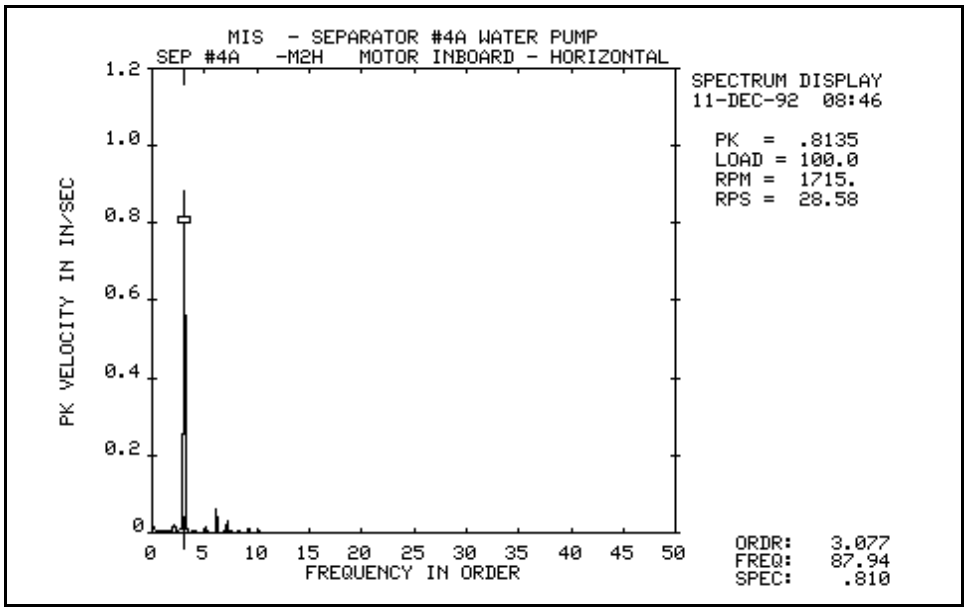

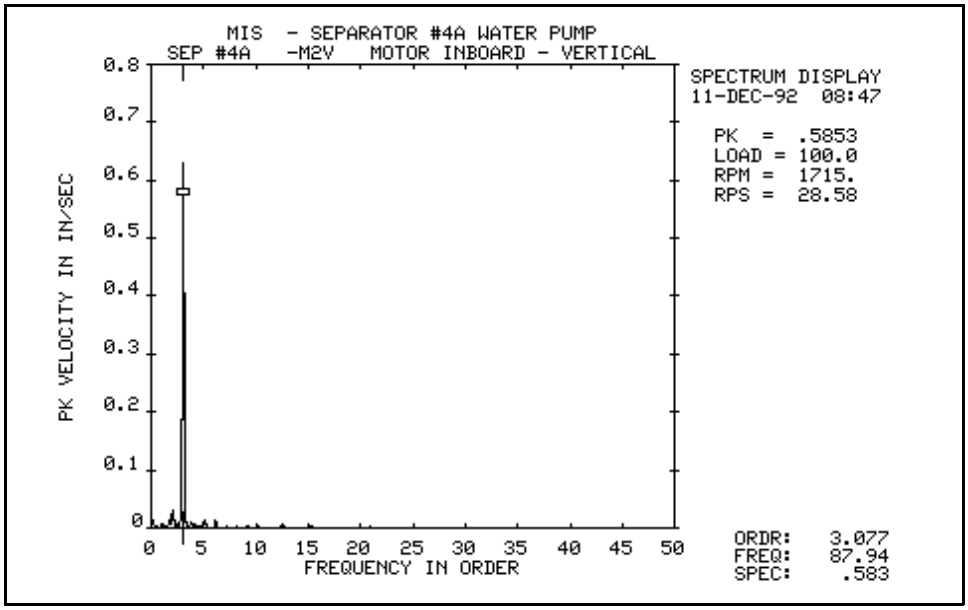

The spectrum for Motor Inboard Horizontal (M2H) and Motor Inboard Vertical (M2V) are shown above. Notice the 3xTS amplitude is consistent between the horizontal and vertical reading.

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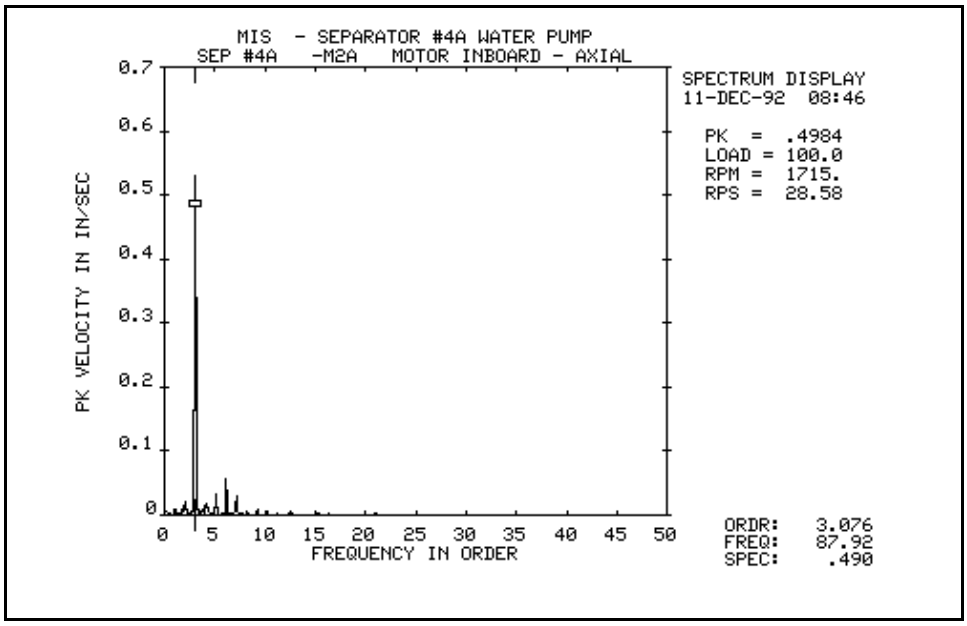

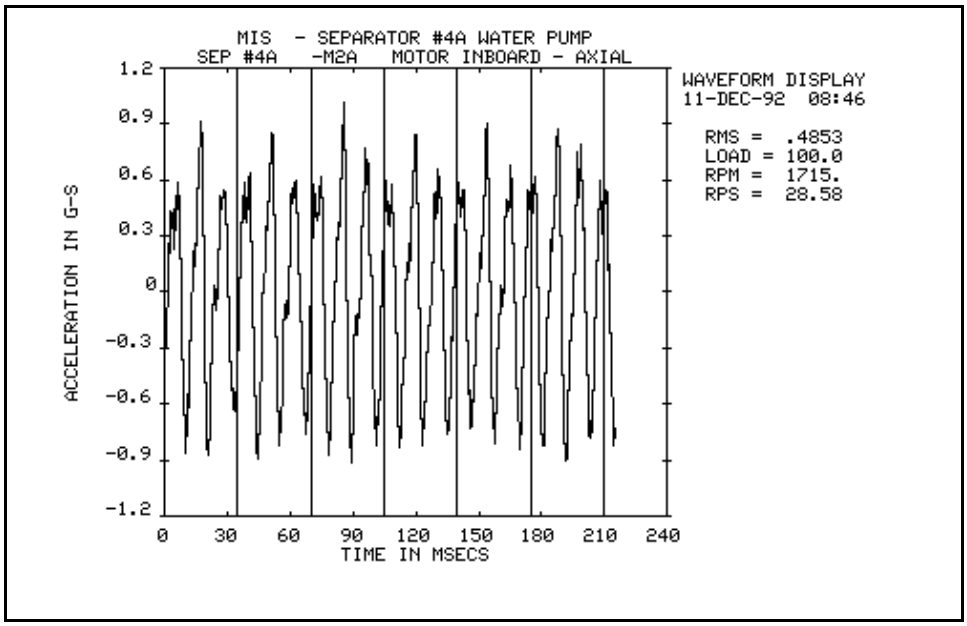

The axial reading also shows the angular misalignment defect. In fact, its amplitude is more than half the amplitude of the radial readings. The waveform shows the sinusoidal pattern with three distinct peaks per one-shaft revolution. This repetitive shape, however, lacks complexity, and the amplitudes of the peaks remain relatively low.

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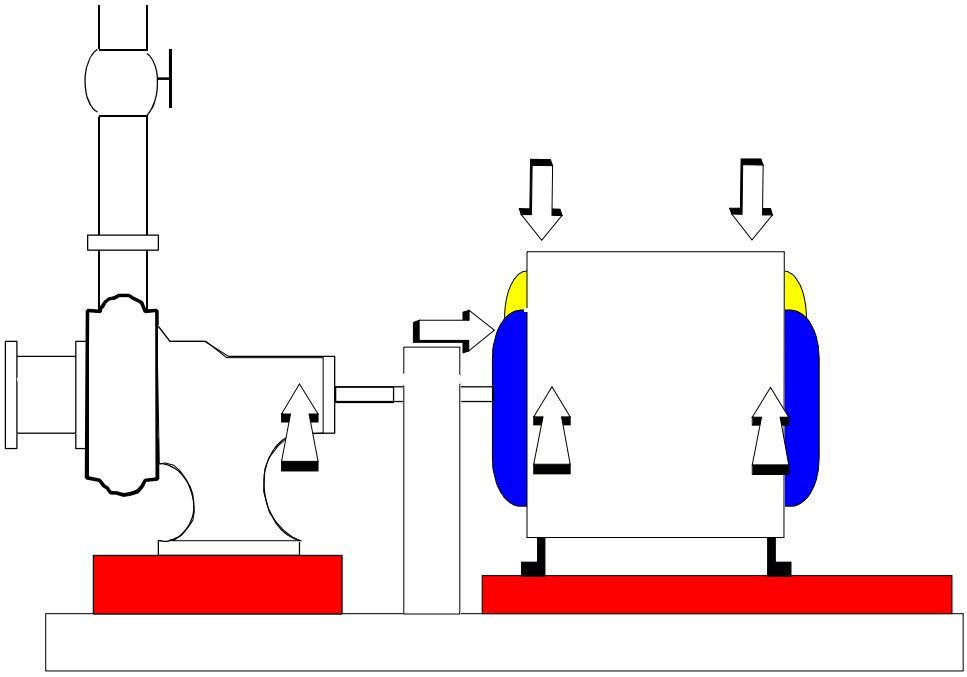
Misalignment Case #5 - Separator Water Pump #2



❶ 15-HP, 1800-RPM AC Induction Motor.

❷ High 3xTS vibration on the motor and pump.

❸ The unit has a three-jaw coupling.

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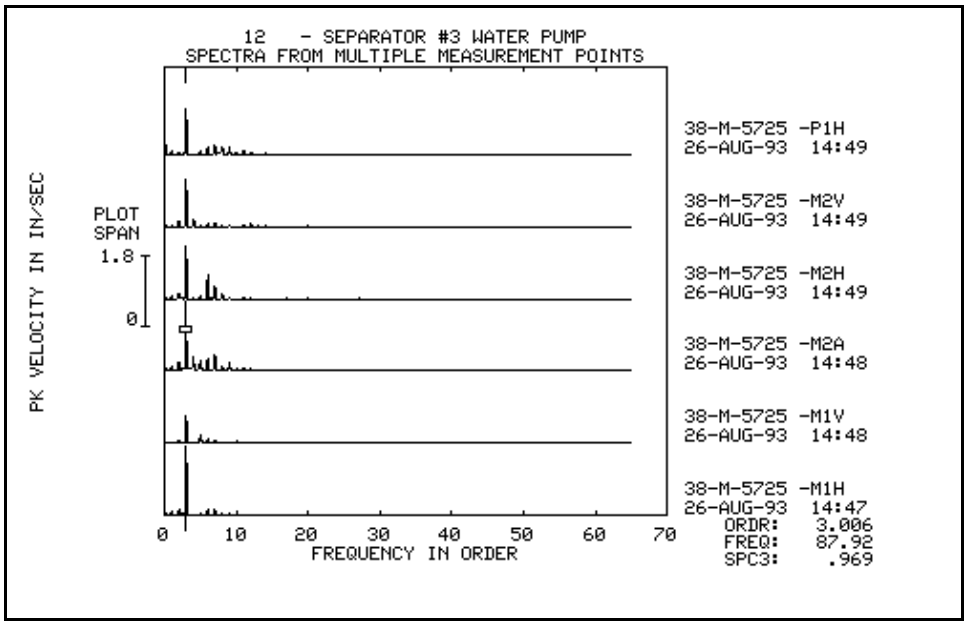
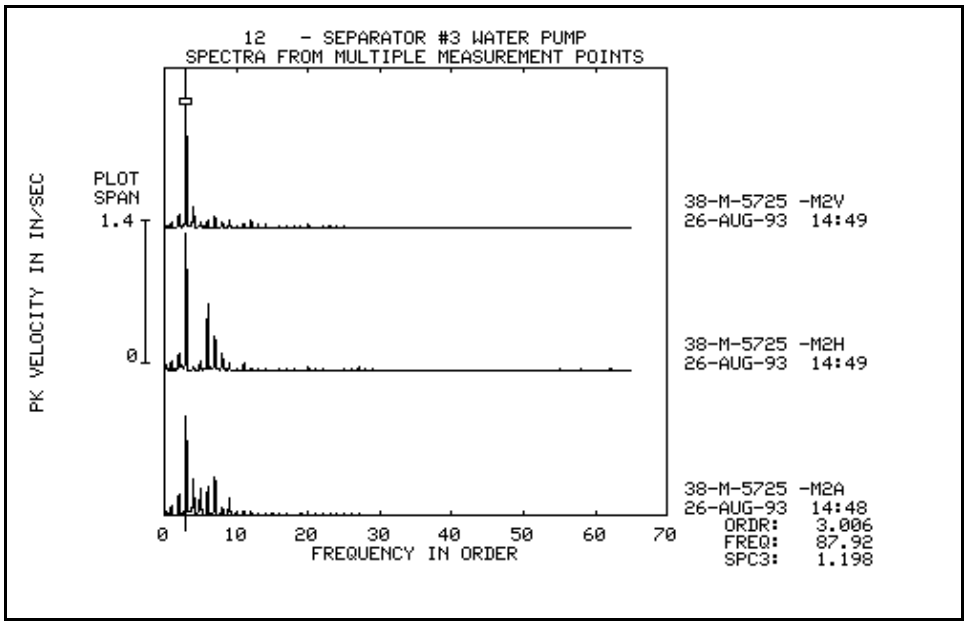


The multiple plot spectrum above shows the measurement points most effected. A cursor marks the 3xTS peak on the motor inboard axial point. Notice the amplitudes from the 3xTS vibration exceeds 1.0 IPS.



The motor inboard measurement points are shown above. Some looseness is now seen in the spectra. The presence of looseness helps distinguish this coupling defect from simple misalignment. The next page shows the motor inboard horizontal spectrum.

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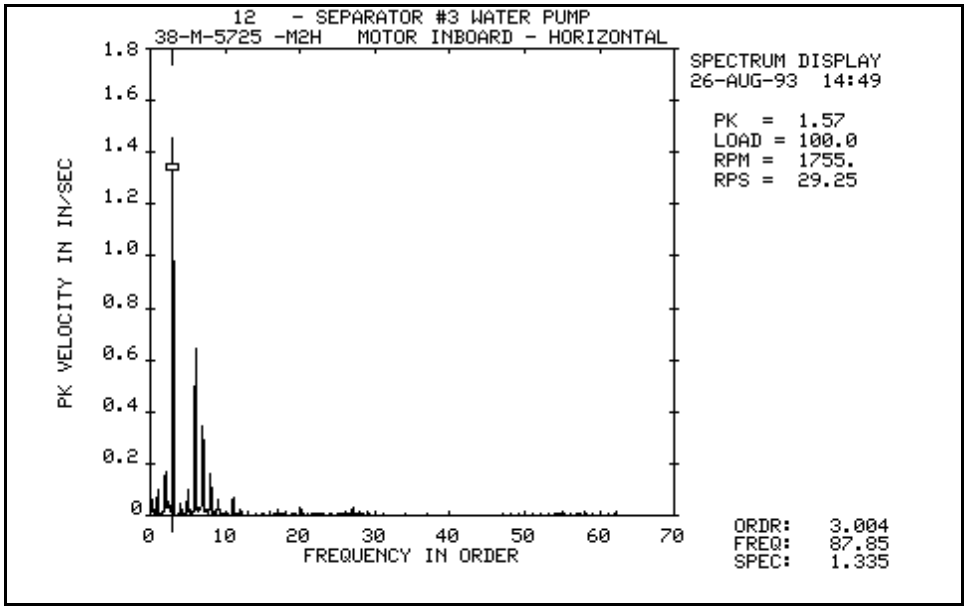
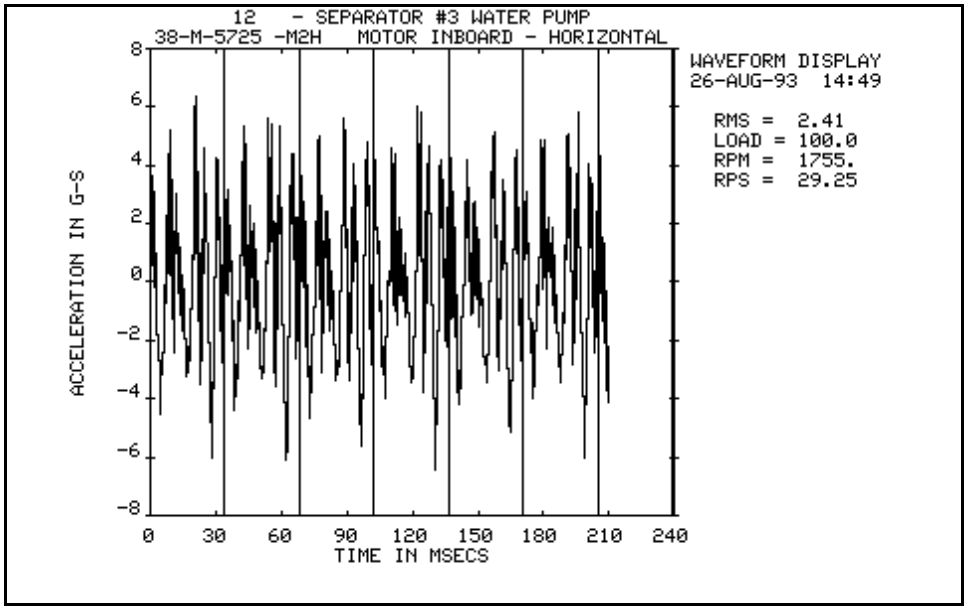


The cursor marks the 3xTS peak measuring an amplitude of over 1.3 IPS. Notice the presence of 3xTS harmonics. The amplitudes of this coupling defect dictates a high priority of concern.



The waveform from the previous spectrum is displayed above. The waveform does show a dominant 3xTS pattern. However, the looseness drives the randomness and complexity.

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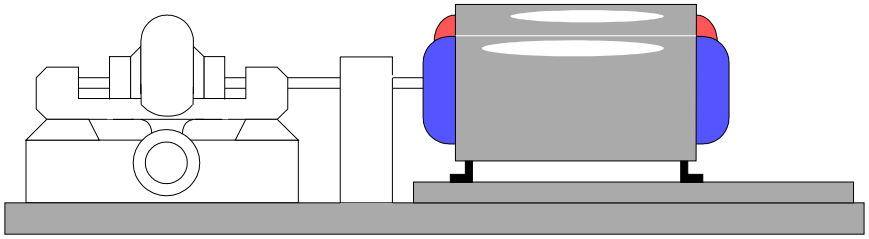
Misalignment Case #6 - Soft Foot on Motor

Speed **3560 RPM**
H.p. **150**



#2 HP Fresh Water Pump

❶ Rolled steel frame motor vs. cast iron

❷ Motor just installed and laser aligned

❸ High 1x RPM in radial directions

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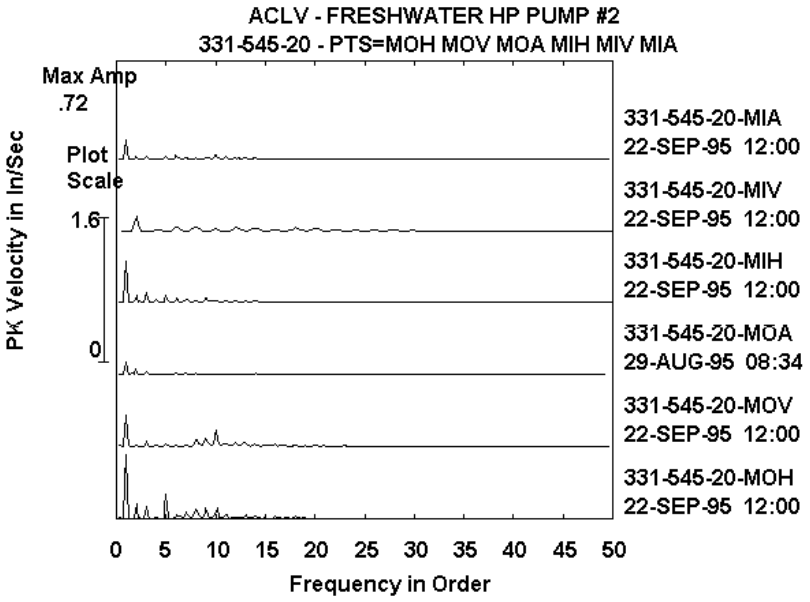
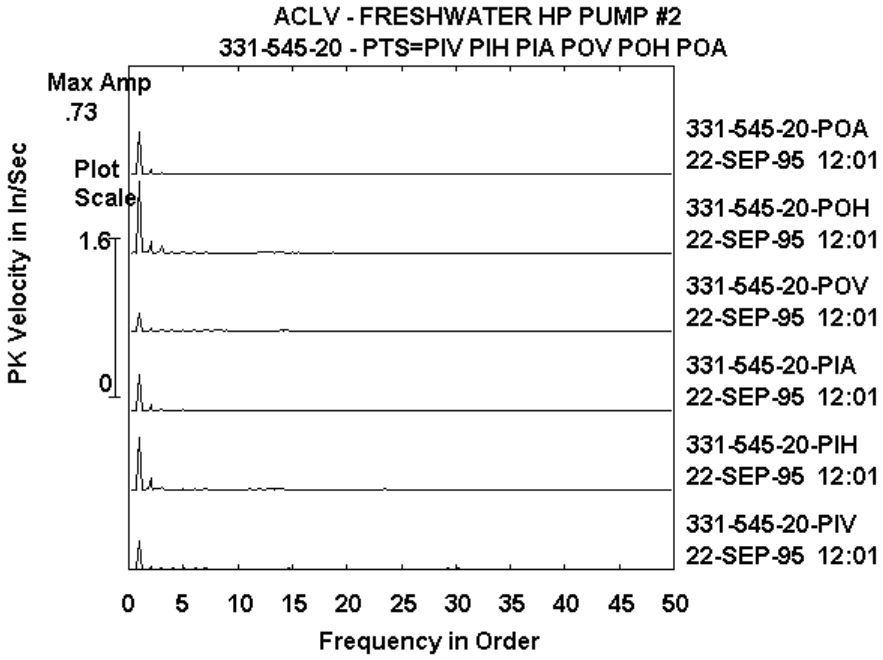


The multi-point plot above shows the motor points. The radial positions show 1x TS dominant with many harmonics.



The pump points also show high radial vibration with 1x TS predominant.

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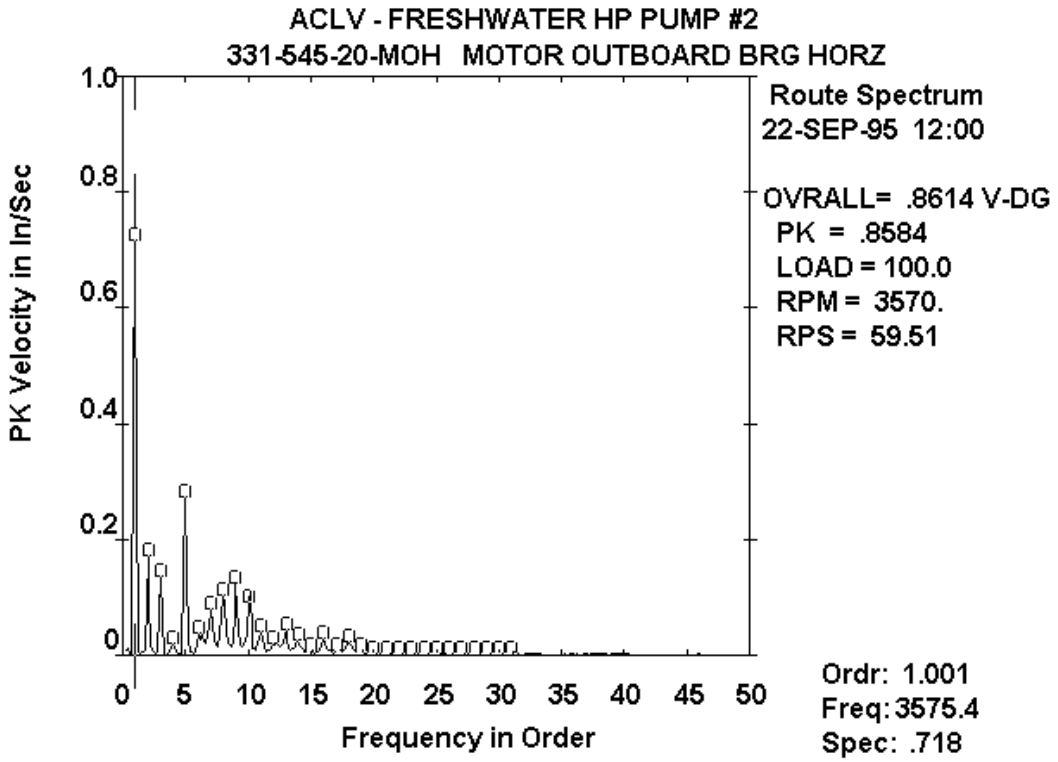

The MOH position shows the severity of the defect. A running soft foot check was performed by using the monitor mode to monitor the spectrum while loosening the motor hold down bolts one at a time. When the front motor bolt was loosened, the vibration dropped to less than .1 in/sec. The soft foot was binding the motor bearings and causing misalignment. The harmonics are the result of forced looseness.

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Section Six
Looseness

Objectives

- Identify looseness in machinery by examining both spectral and waveform data.
- Understand that looseness can include several different variations of mechanical and structural looseness.

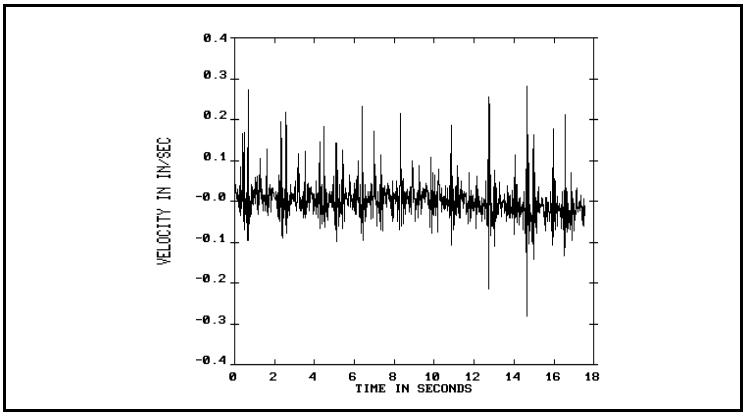
What is Looseness?

Two Types, Structural and Component Looseness

Mechanical looseness occurs when structural or rolling element components do not fit properly. A large number of TS harmonics characterizes looseness in the spectrum. In some cases and stages, looseness exhibits subharmonic multiples of ½xTS. The highest amplitude typically appears radially in the vertical direction. If a coupling is worn, however, axial looseness becomes a possibility.

The waveform shows a great deal of energy and impacting in a random, high frequency pattern. You may also see truncation of the time waveform on rare occasions.

The phase associated with looseness varies from one measurement point to another, and it is unsteady. If the rotor is loose on the shaft, it may slip each time the machine starts, causing the phase to shift from one startup to the next. Mechanical looseness has several stages, as shown on the following page.



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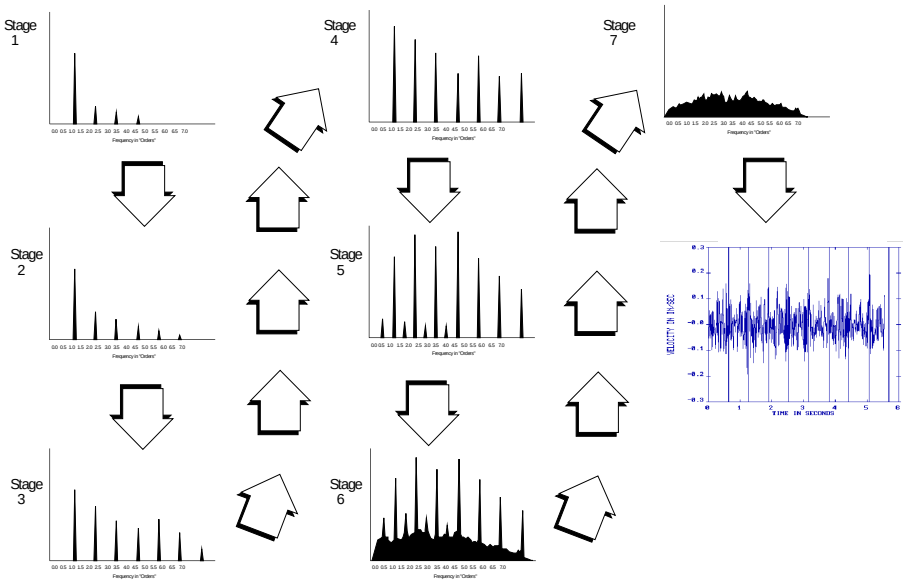
Stages of Looseness



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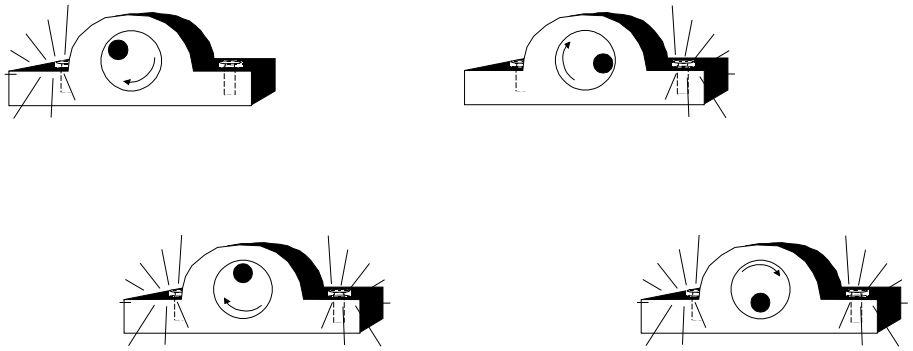
Loose Bearing Housing

Loose bearing housing is caused by looseness of the bolts holding the pillowblock bearing down. This is also called a ***wobble effect*** because of its physical and spectral characteristics. Cracks in the bearing pedestal or frame structure can cause this vibration as well.

The spectrum demonstrates a pattern of synchronous harmonic peaks similar to mechanical looseness. The dominant frequency, however, often appears at 4xTS radially as shown below.

The waveform looks like mechanical looseness because of its random and nonperiodic nature, and it displays significant impacting and energy. The phase is unstable.



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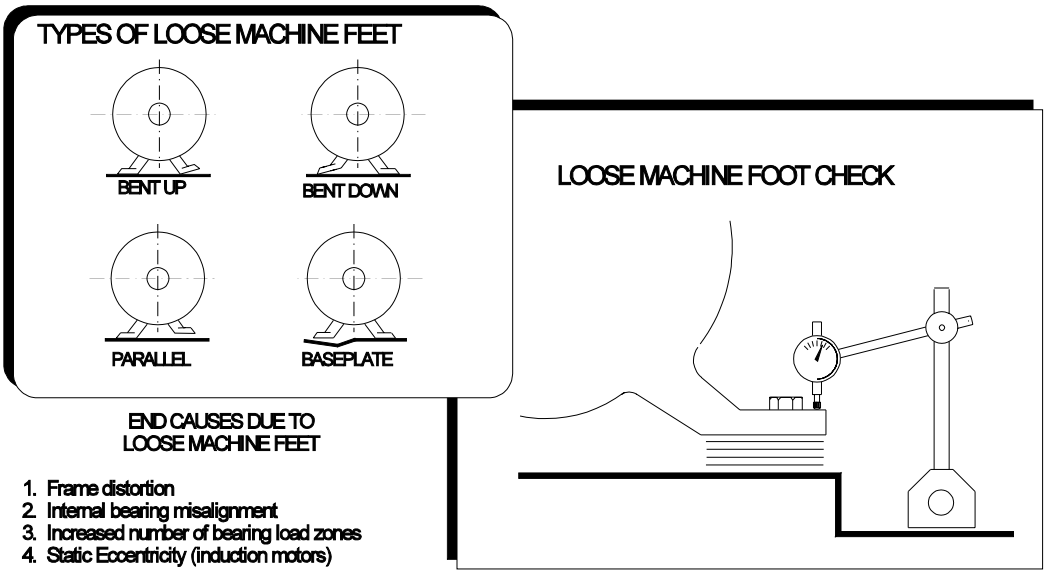

Loose Machine Feet

Loose machine feet, also called soft foot, is caused by a structural weakness or looseness in the foundation, baseplate, or machine feet. The base or frame may have become distorted, grouting may have deteriorated, or the hold-down bolts may have become loose at the base.

The dominant frequency in the spectrum is at 1xTS radially. The waveform is periodic and should demonstrate a strong cycle-per-shaft revolution.

The phase measurement provides the determining factor to distinguish this condition from imbalance and other 1xTS problems. Measure the phase of the frame or base, baseplate and machine foot vertically. If the phase shifts 180° between those measurements, soft foot is the source.

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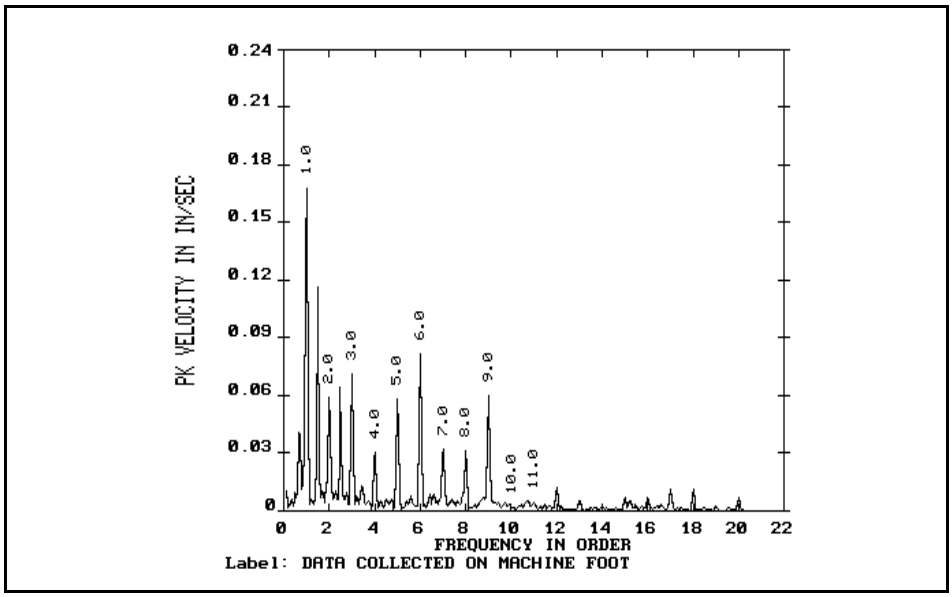

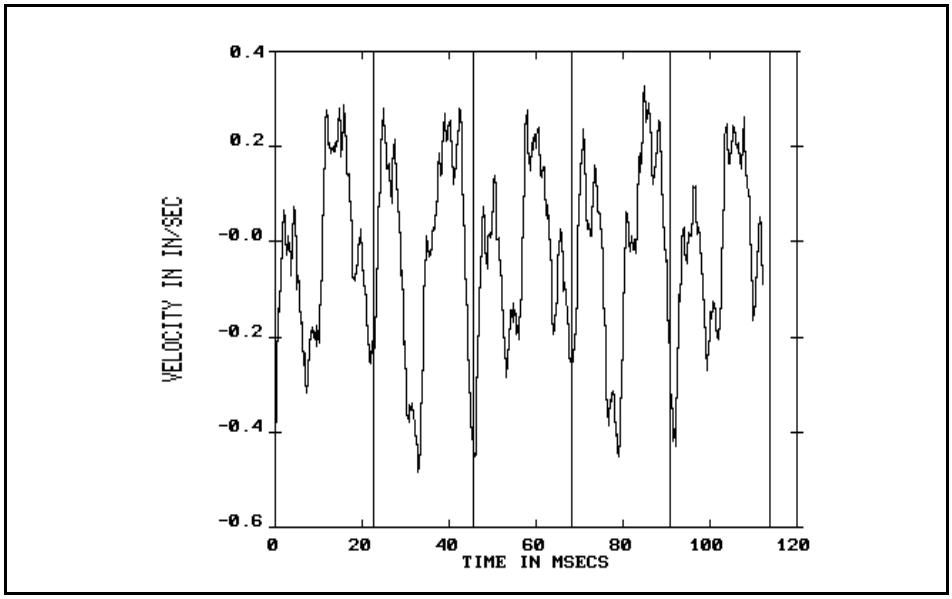

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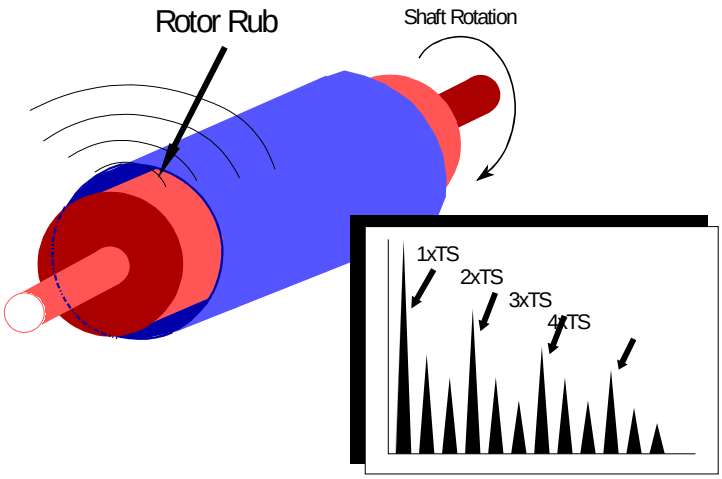

Rotor Rub

Rotor rub is often a significant problem in rotating machinery and is frequently caused by Looseness or Misalignment. Rubbing occurs when rotating elements come in contact with stationary components of a machine.

The pattern in the spectrum demonstrates fractional subharmonics of TS radially, usually harmonics of ⅓xTS. The nature of rubbing and impacting components would explain the common excitation of rotor and/or component natural frequencies.

The pattern in the time waveform is quite distinct. The waveform appears truncated or flattened on one side. The truncation demonstrates the lack of freedom of movement incurred by the rotor as it rubs the stationary component. The phase shows constant change. Whether or not the rotor rubs throughout the entire revolution or just part of it determines how significantly the phase changes.

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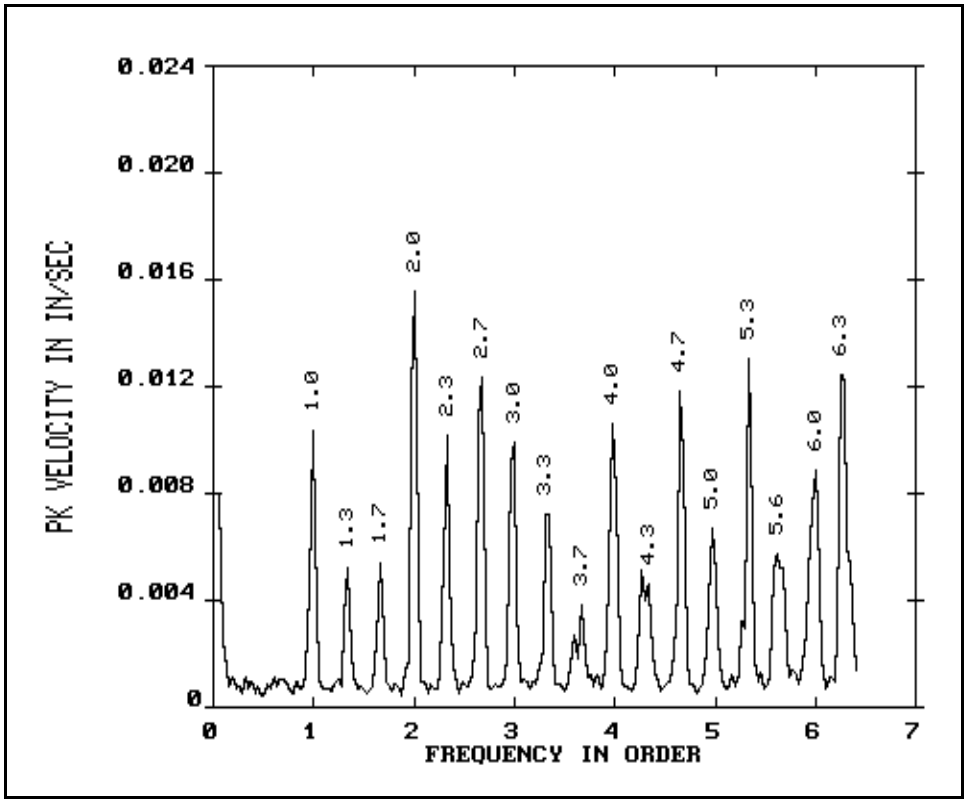

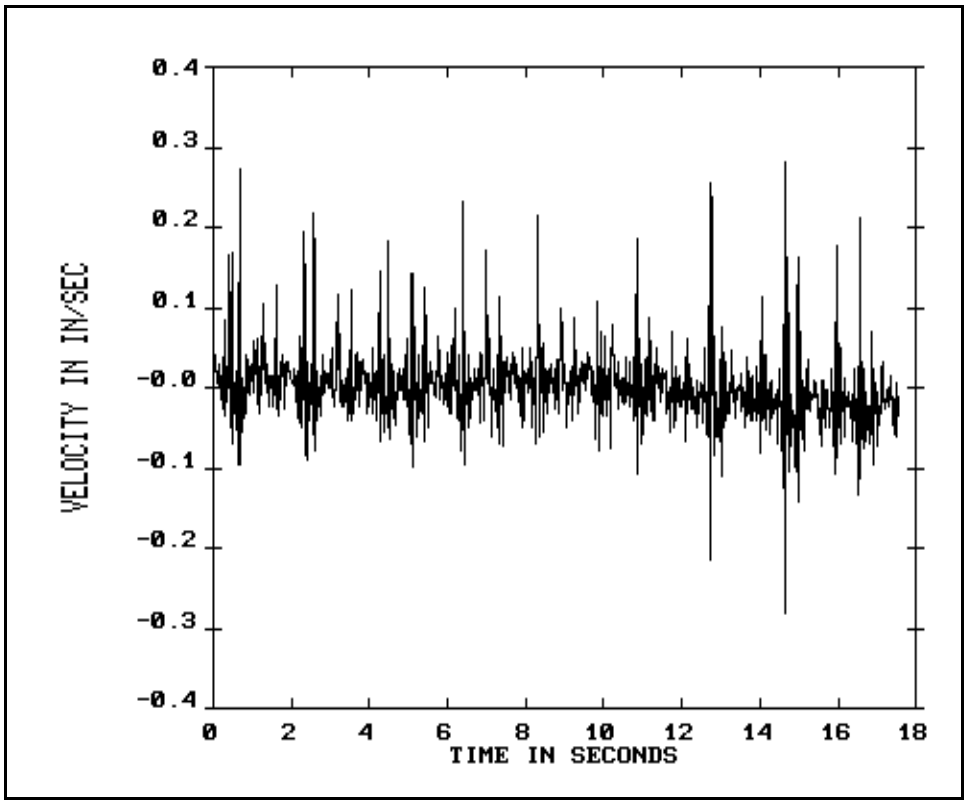

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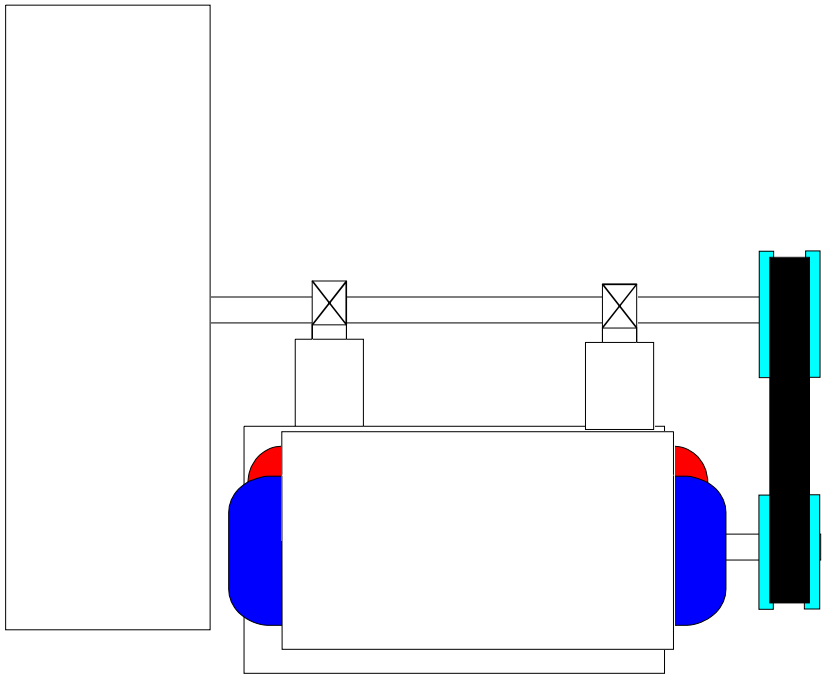
Looseness Case Histories

Looseness Case #1 - Stack Fan



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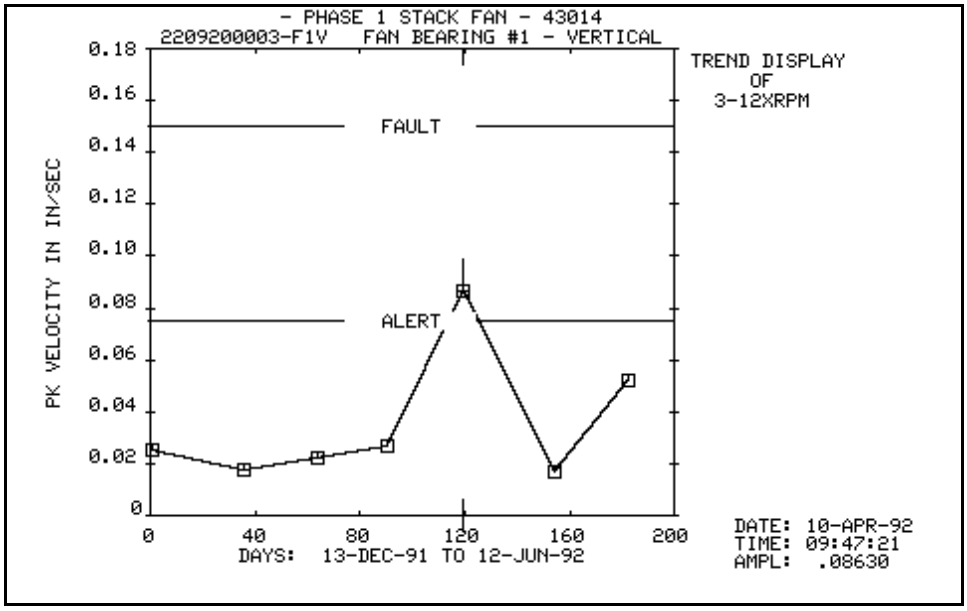
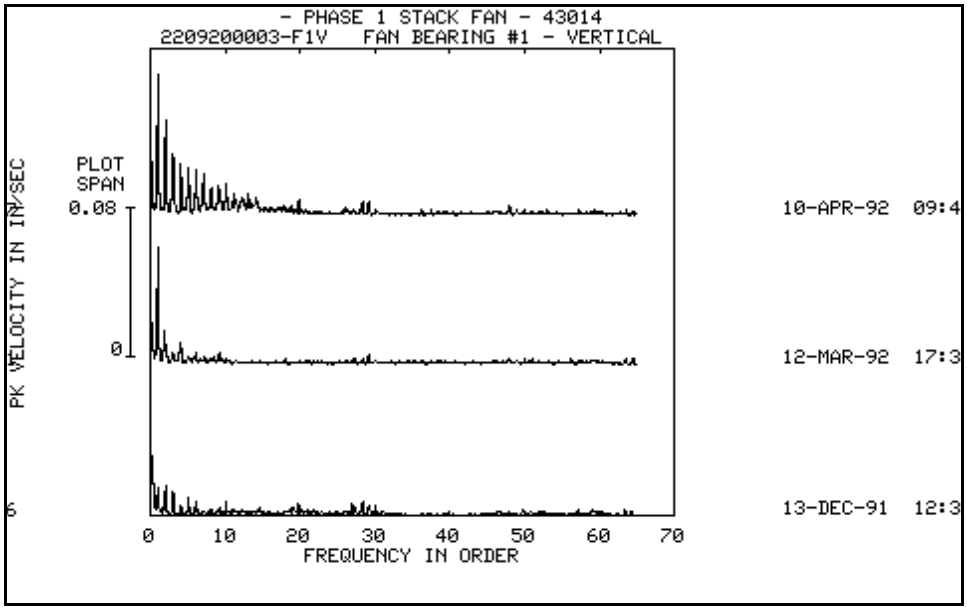


This data shows a trend plot for a fan inboard bearing from the 3 to 12 xTS band. Notice the plot shows an increase for the data collected on 10-APR-93. The figure below shows a multiple plot for the previous month's data.



The multiple-spectrum plot for the fan point F1V is displayed. The data shows an increase in vibration energy over the three collection periods. Notice the 1xTS peak begins to increase in amplitude in March.

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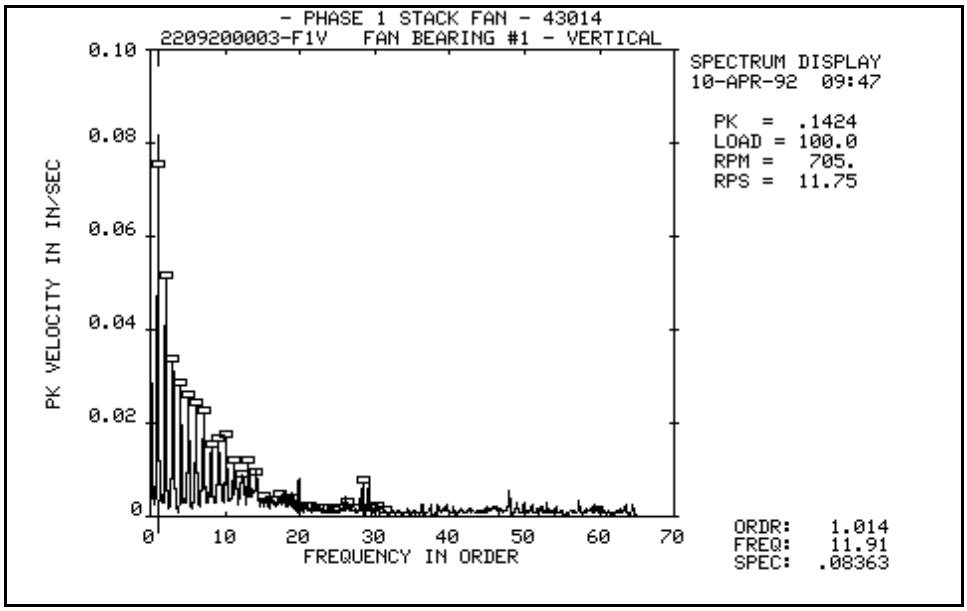
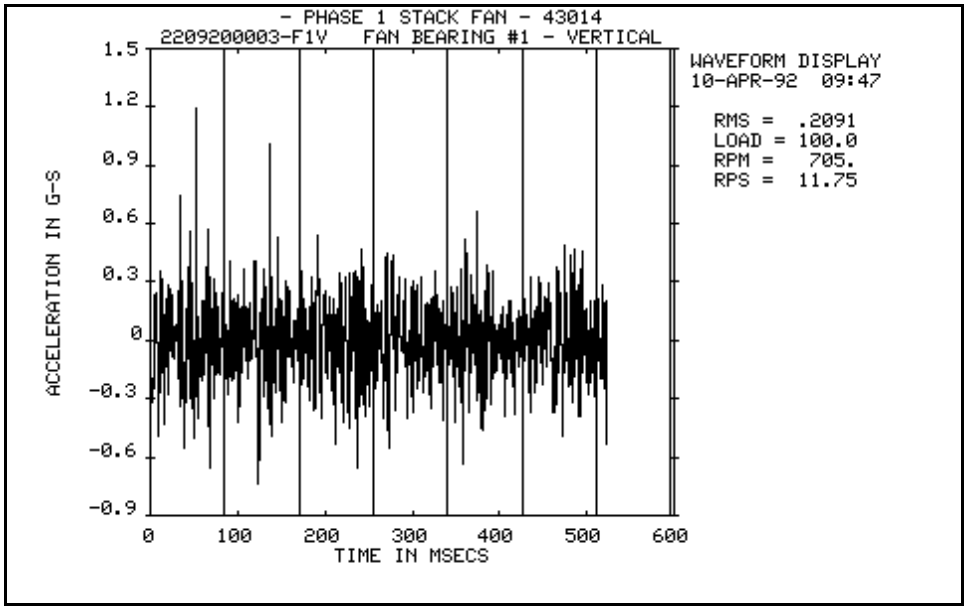


The plot above shows the April spectrum for the fan inboard bearing. This plot shows many harmonics of shaft turning speed. These harmonics are whole number multiples of shaft turning speed and are classified as synchronous peaks. The data is an illustration of a looseness problem.



The waveform data for the F1V measurement point is displayed above. Notice the energy level is low, only 0.2 g's. The low amplitudes and little impacting indicate that the looseness is not yet severe. The typical looseness waveform is shown above by its erratic and nonrepetitive shape.

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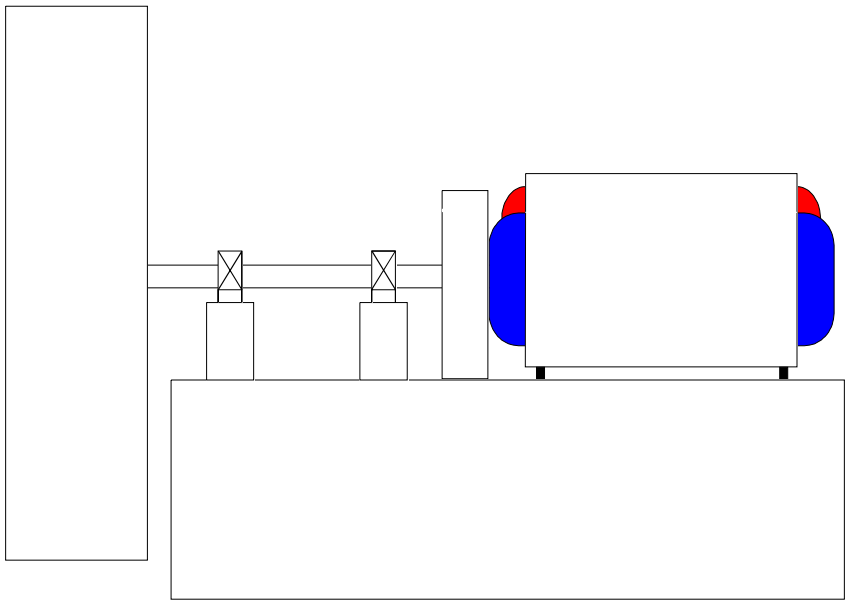
Looseness Case #2 - Combustion Air Fan



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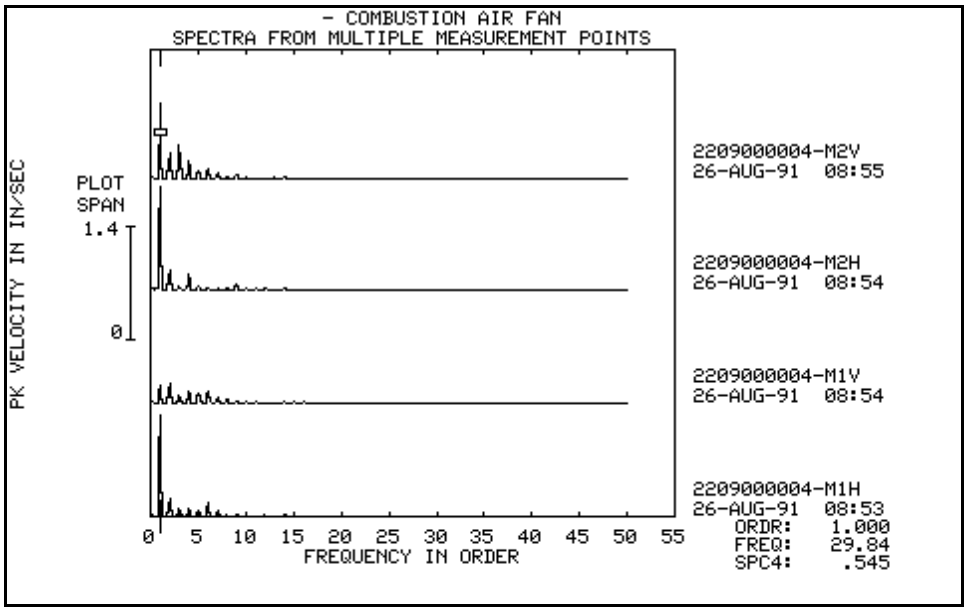
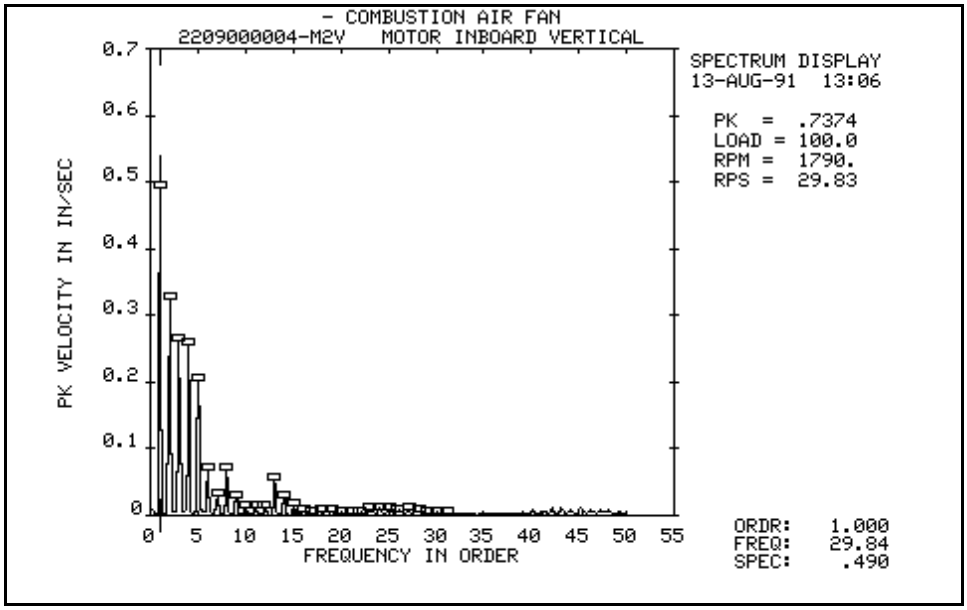


The multiple-points plot shows the spectra for the radial motor points. The cursor marks the 1xTS peak on the motor inboard vertical (M2V) measurement point. Most of the energy appears to be near motor turning speed.



The single spectrum showing measurement point M2V is displayed above. The cursor marks the motor running speed at 29.84 Hz or 1 order. The harmonic cursor marks multiples of turning speed past 15 orders. This spectral pattern signifies machine or component looseness.

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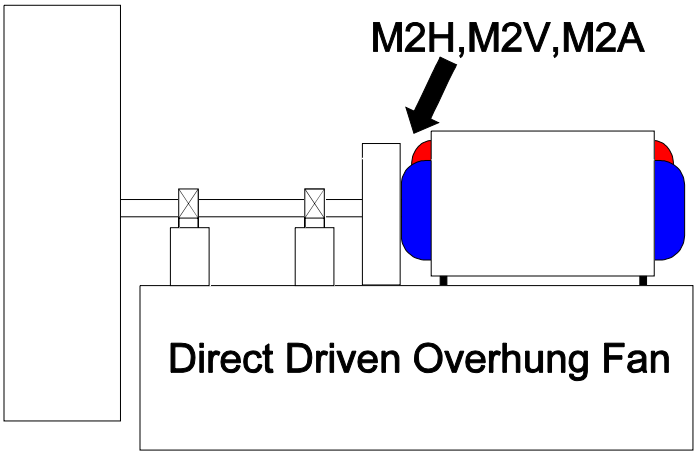
Looseness Case #3



❶ 200-HP, 1200-RPM motor.

❷ Overhung fan, direct driven.

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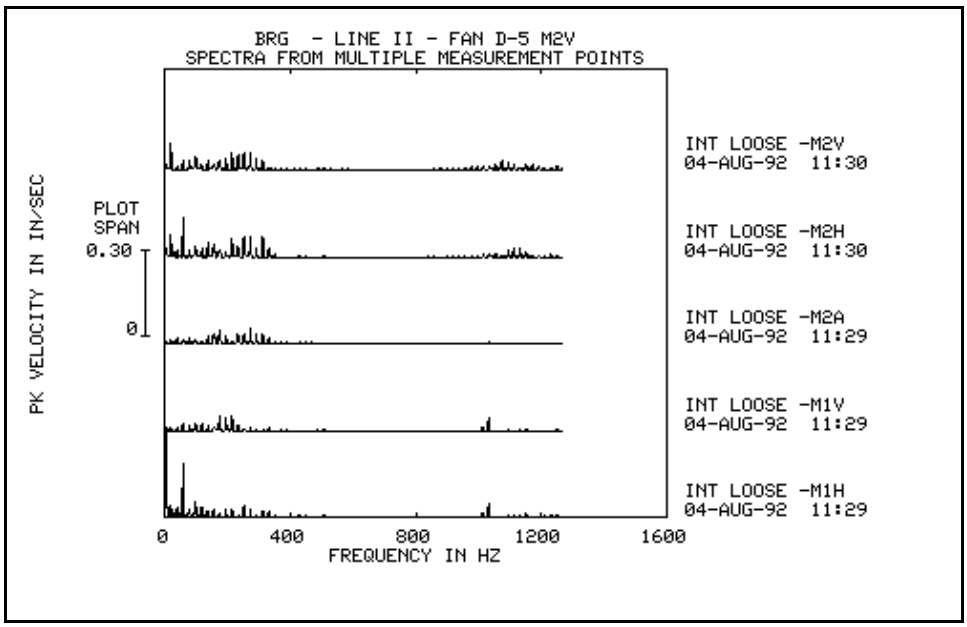
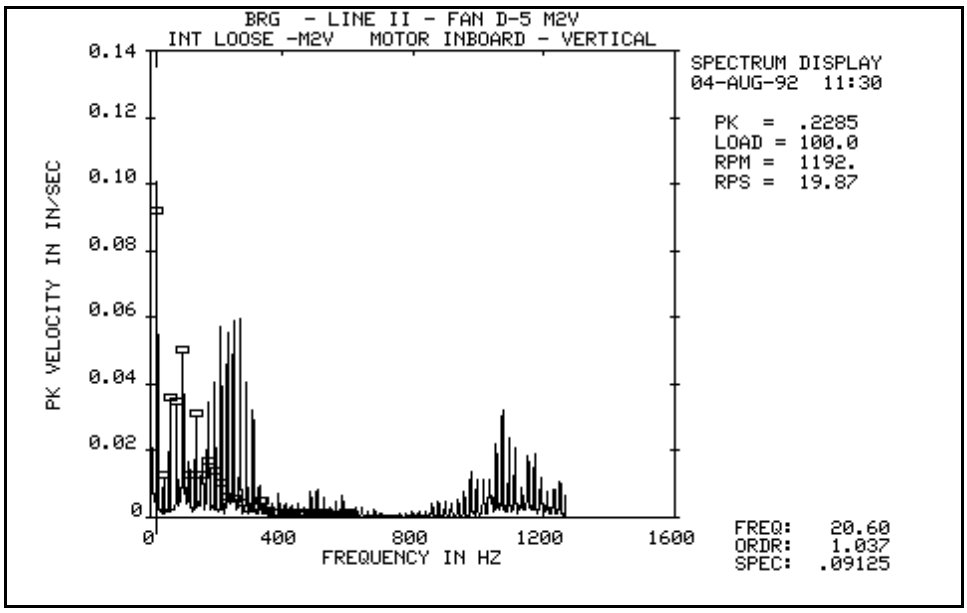


The above multiple-point plot shows all the motor points. Most of the activity is occurring at the inboard points.



The above single spectrum is shown for point motor inboard vertical (M2V). Several multiples of turning speed are identified by the harmonic cursor. This is an indication of looseness.

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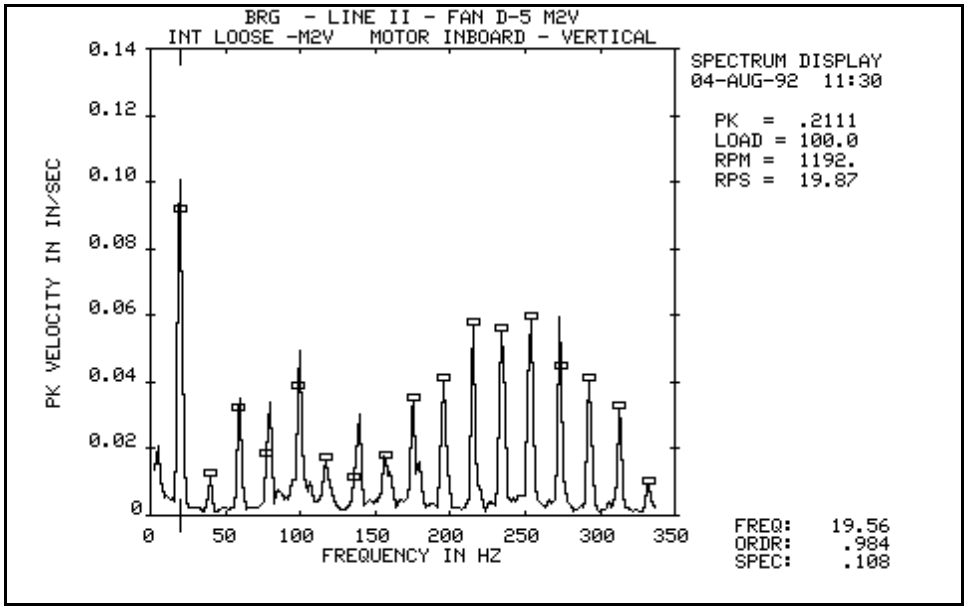
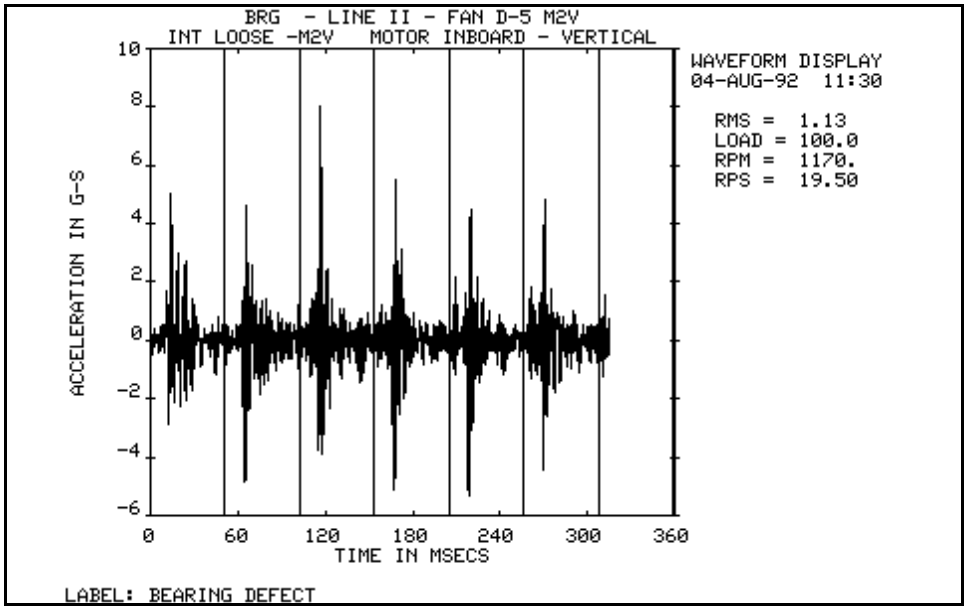


The above spectrum is an expanded plot of the previous spectrum. Notice the pattern shows an increase in amplitude of the harmonic peaks as you move away from shaft turning speed.



The waveform shows the impacting and ringing down pattern. Also, notice the level of g's indicated in the waveform. Why is the waveform showing this pattern instead of a random looseness pattern? This is an indication of **bearing looseness** showing impacting in the waveform, as opposed to structural looseness. **Structural Looseness** typically shows more of a **random pattern** in the waveform.

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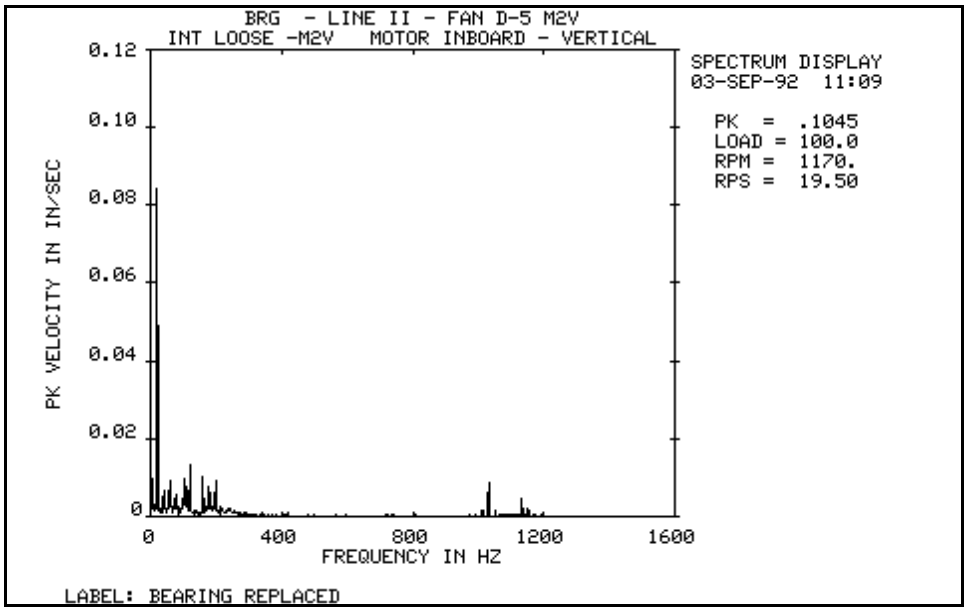
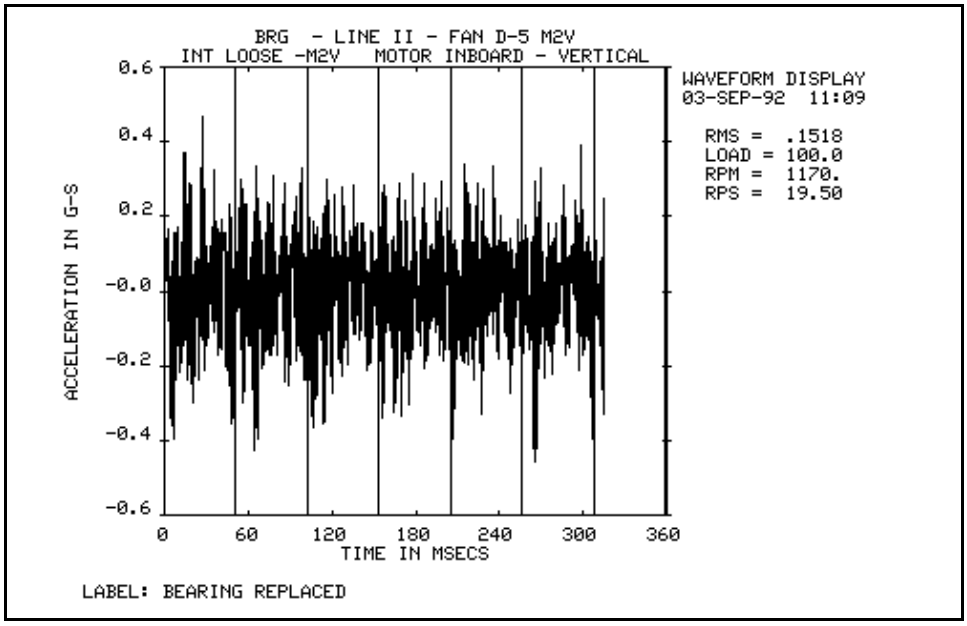


The above spectrum was taken after the bearing was replaced. Notice the drop in overall energy as well as the decrease in harmonic activity.



The waveform is showing very little activity after the bearing was replaced. The impacting and ringing down pattern has disappeared.

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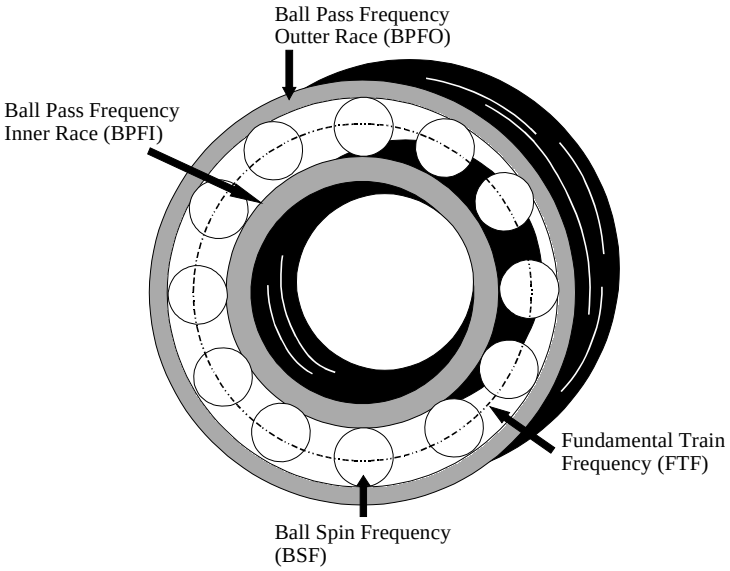
Section Seven
Antifriction Bearings

Objectives

- Calculate the rolling element bearing defect frequencies.
- Recognize bearing defect frequencies in spectral data.
- Use the time waveform when diagnosing bearing defects.
- Determine defect severity based on spectral frequency patterns instead of spectral amplitude alone.

Introduction

Rolling element bearings have specific bearing failure modes that can be observed in the spectral and the waveform data. Bearing frequencies are different than some other vibration frequencies present in a frequency spectrum in that bearing frequencies are truly defect frequencies. These frequencies will not appear unless the bearings are actually defective. Each failure mode centers around a particular defect site and consists of defect frequencies that appear because of defects in the bearing components.



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The four primary rolling element defect frequencies are classified by their location.

- FTF (Fundamental Train Frequency) or Cage Frequency
- BSF (Ball Spin Frequency)
- BPFO (Ball Pass Frequency Outer Race)
- BPFI (Ball Pass Frequency Inner Race)

Note: The BPFI will always be the highest calculated frequency for rolling element bearings.

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Displayed below is a page from the CSI bearing file.

- The BPFI is the highest calculated bearing frequency for each bearing.
- A rule to apply when selecting an F_{max} on machines with rolling element bearings is:

$$\text{BPFI} \times \text{\# of balls/rollers} = F_{max}$$

DIRECTORY OF BEARING FILE: CSIBRG.DAT

BRG ID	BEARING TYPE	#B/R	FTF	BSF	BPFO	BPFI
1	SKF 1111	9	.402	2.461	3.621	5.379
2	SKF N206	12	.419	2.988	5.023	6.977
3	SKF N207	12	.415	2.860	4.981	7.019
5	SKF N209	14	.423	3.170	5.922	8.078
6	SKF N210	15	.429	3.429	6.429	8.571
7	SKF N211	15	.429	3.452	6.436	8.564
8	SKF N212	15	.430	3.492	6.447	8.553
9	SKF N213	15	.430	3.507	6.452	8.548
10	SKF N214	16	.433	3.683	6.933	9.067
11	SKF N215	17	.432	3.593	7.339	9.661
12	SKF N216	16	.432	3.602	6.910	9.090
13	SKF N217	16	.432	3.620	6.915	9.085
14	SKF N218	16	.428	3.398	6.847	9.153
15	SKF N219	16	.428	3.415	6.853	9.147
16	SKF N220	16	.429	3.431	6.858	9.142
17	SKF N222	16	.426	3.297	6.814	9.186
18	SKF N224	16	.428	3.417	6.854	9.146
19	SKF N225	18	.433	3.683	7.800	10.200
20	SKF N228	18	.433	3.682	7.800	10.200
21	SKF N305	10	.398	2.342	3.977	6.023
23	SKF N307	11	.404	2.504	4.442	6.558
24	SKF N308	12	.408	2.638	4.901	7.099
25	SKF N309	11	.403	2.493	4.438	6.562
26	SKF N310	12	.406	2.576	4.876	7.124
27	SKF N311	11	.403	2.478	4.432	6.568

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Calculation of Fundamental Rolling Element Bearing Defect Frequencies

Case I Inner race is rotating and outer race is stationary (most common industrial application).

$$FTF = \frac{S}{2} \cdot (1 - \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BPFI = \frac{Nb}{2} \cdot S \cdot (1 + \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BPFO = \frac{Nb}{2} \cdot S \cdot (1 - \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BSF = \frac{Pd}{2Bd} \cdot S \cdot [1 - (\frac{Bd}{Pd})^2 \cdot (\cos\theta)^2]$$

where:

RPM	=	revolutions per minute
S	=	speed, revolutions per second
FTF	=	fundamental train (cage) frequency
BPFI	=	ball pass frequency of the inner race
BPFO	=	ball pass frequency of the outer race
BSF	=	ball spin frequency
Bd	=	ball or roller diameter
Nb	=	number of balls or rollers
Pd	=	pitch diameter
Θ	=	contact angle

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Case II Inner race is stationary and outer race is rotating (e.g., front wheels of some cars).

$$FTF = \frac{S}{2} \cdot (1 + \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BPFI = \frac{Nb}{2} \cdot S \cdot (1 + \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BPFO = \frac{Nb}{2} \cdot S \cdot (1 - \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BSF = \frac{Pd}{2Bd} \cdot S \cdot [1 + (\frac{Bd}{Pd})^2 \cdot (\cos\theta)^2]$$

where:

RPM	=	revolutions per minute
S	=	speed, revolutions per second
FTF	=	fundamental train (cage) frequency
BPFI	=	ball pass frequency of the inner race
BPFO	=	ball pass frequency of the outer race
BSF	=	ball spin frequency
Bd	=	ball or roller diameter
Nb	=	number of balls or rollers
Pd	=	pitch diameter
Θ	=	contact angle

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Two other relationships are also important:

❶ The sum of the BPFO and the BPFI is equal to the number of rolling elements multiplied by the shaft turning speed.

$$BPFO + BPFI = \#\ balls * TS$$

❷ For the Inner Race rotating, the FTF is equal to the BPFO divided by the number of rolling elements.

$$FTF = BPFO / \#\ balls$$

❸ For the Outer Race rotating, the FTF is equal to the BPFI divided by the number of rolling elements.

$$FTF = BPFI / \#\ balls$$

Obviously, each different bearing type has different bearing defect frequencies. The frequencies depend on the bearing geometry. These frequencies should not appear in the spectral and waveform data unless a defect actually exists.

The defect frequency amplitudes depend on the defect severity and the bearing failure mode.

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Vibration Data

Spectral Data

Measuring the ***spectral*** data can detect these defect frequencies. Not only are the primary defect frequencies of interest when diagnosing bearing failures, but also the frequency harmonics. The maximum analysis frequency (F_{max}) during data collection should be high enough to capture the higher frequency harmonics. An F_{max} set at or above a frequency equal to the BPFI multiplied by the number of rolling elements allows all the bearing defect frequencies to be captured. It is the detection of the higher frequency harmonics of the BPFI and the BPFO defect frequencies that often appear during the first failure stages of bearing failure. It is the initial detection of these defect frequencies that allows an analyst to track the bearing defect.

Each of the four primary bearing defects can be considered to be a particular mode of failure.

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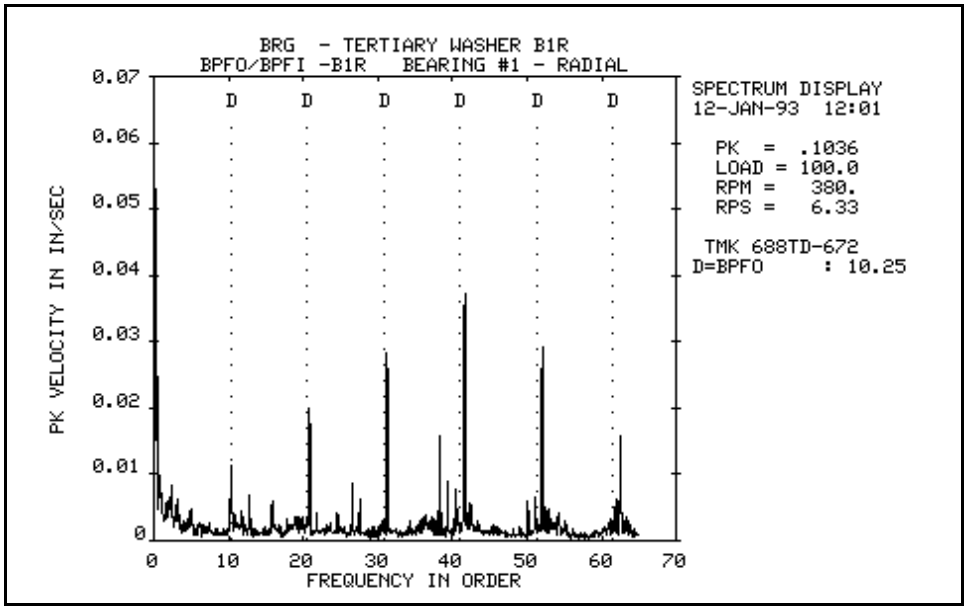
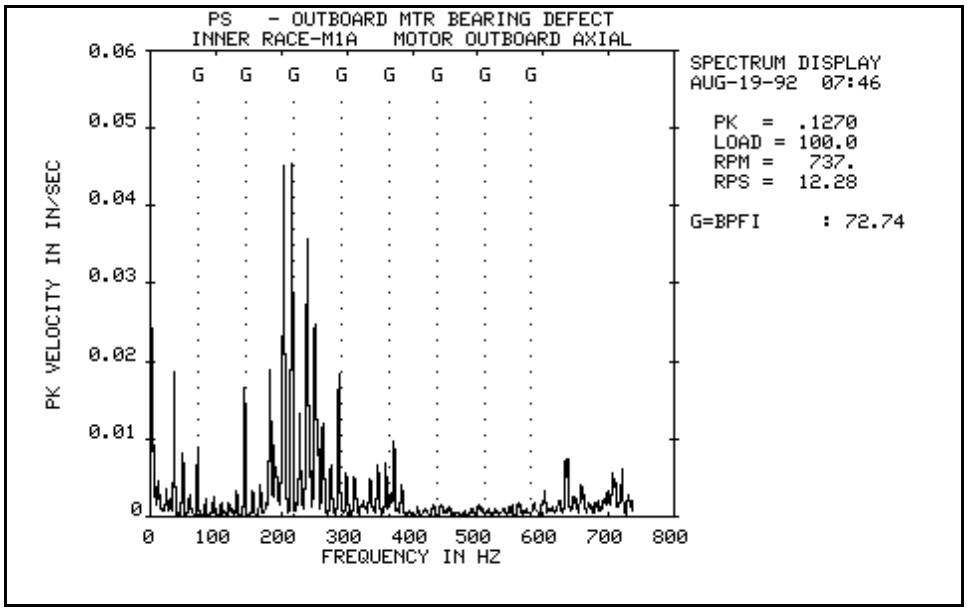
Ball Pass Frequency Outer Race Defect



The spectrum above shows a bearing with outer race defect frequencies marked.

Ball Pass Frequency Inner Race Defect



The spectral data above is from a bearing with inner race defect frequencies marked.

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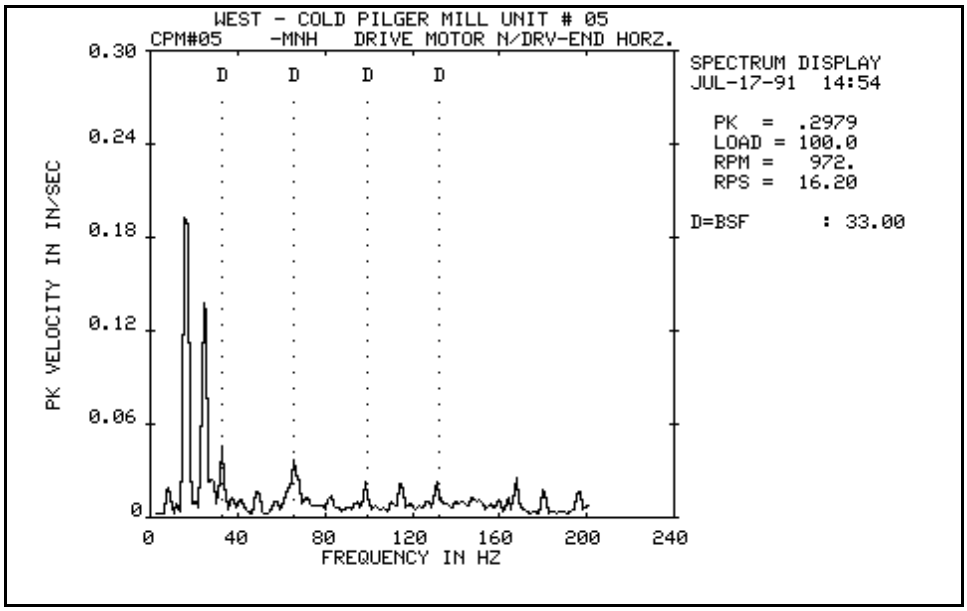
Rolling Element Defect (Ball Spin Frequency)



The spectrum above shows defect frequencies related to a rolling element defect.

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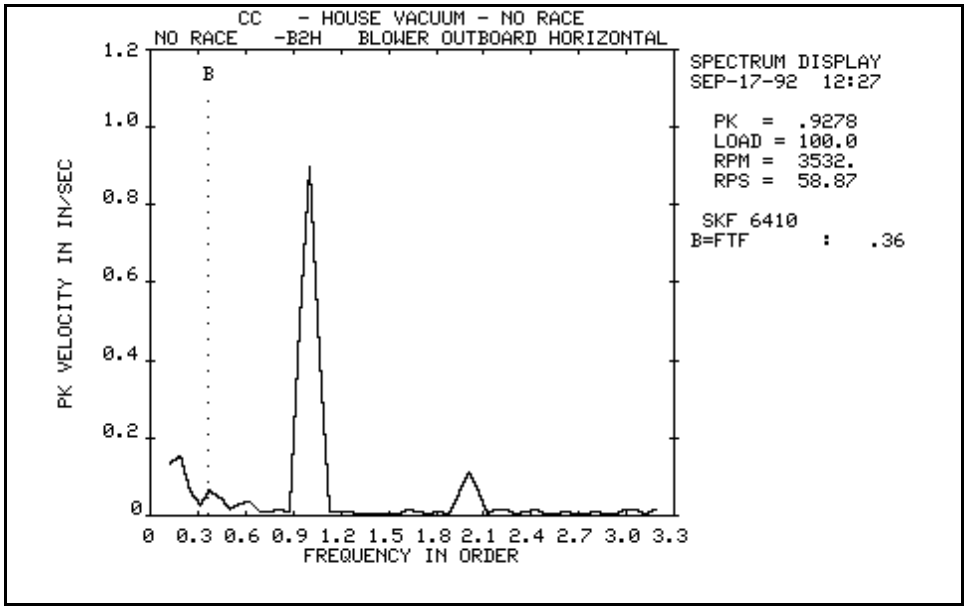
Cage Defect



The spectrum shown above shows the cage defect frequency appearing at less than 1 x turn speed. At about .3 xTS. This is normally where a cage defect appears on a new bearing with a cage defect.

On a bearing that has been in service for some time a cage defect normally appears as a sideband about a BPFO or a BPFI defect frequency.

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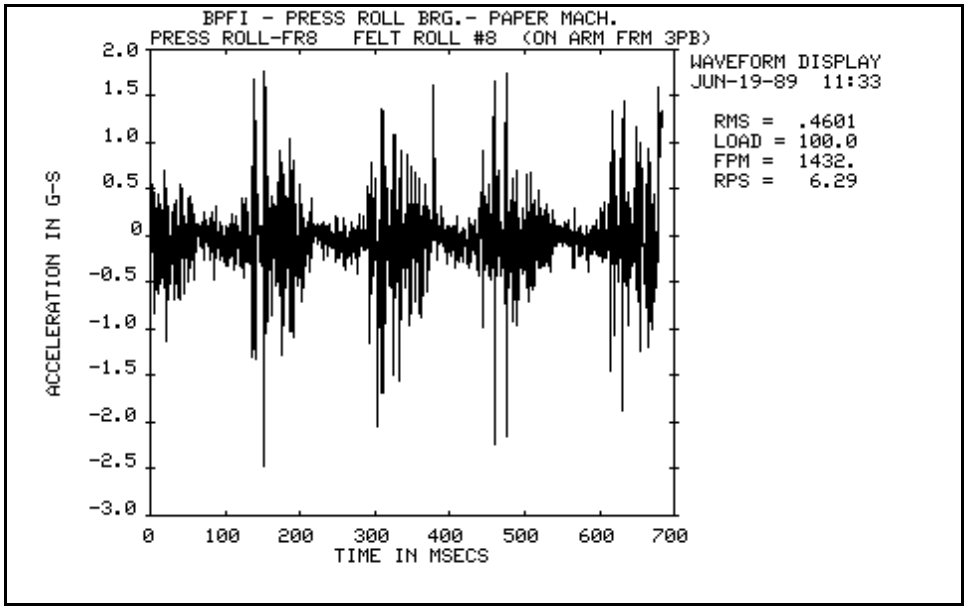
Time Waveform

The time waveform should not be neglected when analyzing rolling element bearing defects. The waveform amplitude pattern assists the analyst in confirming some bearing defects. The time waveform shows an impact and ring down pattern as the rolling elements pass a defect site.



The waveform above shows classic impact and ring down due to the rolling element impacting the defect site.

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Understanding Amplitude Levels

It is helpful to know what bearing types are in the machines in order to more accurately identify the bearing fault frequencies displayed in the data. However, once the defects are identified, it is not always possible to only trend the amplitude change over time. Some bearings may fail when no defect frequency exceeds 0.07 IPS, while others may continue to run at amplitudes of 0.25 or more IPS.

One reason for amplitude variations is the vibration energy transmission path. Amplitudes for the same defect severity will be lower if the vibration has traveled through several feet of the machine than if the sensor were mounted just a few inches from the bearing. Also, inner race defects may appear at lower amplitudes than outer race defects due to the different transmission path.

Another reason that amplitudes alone cannot be relied upon to provide accurate defect severity is the differences in each machine's mass, stiffness, and damping properties. Even side-by-side machines with the same basic design could have slightly different system properties that could affect the measured amplitudes of similar sized bearing defects on each machine.

For example, the lower frequency amplitudes are affected primarily by the machine's mass. As frequencies increase, the machine's mass has a smaller effect on the vibration and the machine's stiffness begins to have a controlling effect on the vibration amplitude. Therefore, at the lower frequencies closer to turning speed, Sir Isaac Newton's second law is more applicable. This law states that a force is equal to its mass multiplied by the acceleration of the mass. The equation is written:

$$F = M * A$$

For vibration analysis, we can use this equation in a slightly different form to better understand one of the reasons bearings fail at different amplitudes for different machines. This equation can be rearranged:

$$F / M = A$$

Examining this equation, assume the same force is present on two different machines. Machine B is 10 times the mass of Machine A. If the equation is rewritten:

$$M_A * A_A = F_{A,B} = M_B * A_B$$

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Since $M_B = 10\ M_A$, the equation can be rewritten:

$$M_A * A_A = 10M_A * A_B$$
$$(M_A * A_A) / 10M_A = A_B$$
$$(1 / 10)\ A_A = A_B$$

Machine B's acceleration level will be one-tenth of Machine A if all other factors, including the force are held constant.

Bearing Life

The life of the bearing is inversely proportional to the load cubed and the shaft turning speed based on the following equation:

$$H = (C^3 / L^3) * (16667 / RPM)$$

Where,

H	=	life in hours
C	=	bearing capacity in pounds
L	=	bearing load in pounds
RPM	=	shaft turning speed

An increase in the vibration levels increases the effective load, which therefore, decreases the life of the bearing. Increasing the speed decreases the bearing life. The increased shaft speed raises the bearing defect frequencies which results in higher vibration amplitudes as the acceleration levels are amplified at the higher frequencies. This results in higher forces experienced by the bearing since the force is related to the mass and acceleration as discussed previously.

Slower shaft speeds have the opposite effect on bearing life. The faults develop over longer time periods at lower amplitudes.

This is why vibration alarm levels that are adequate for some machines do not work well for other machines. Also, this is the reason that some machines fail at low amplitude defects and other machines tolerate higher amplitude defect related vibration for the same level of defect severity.

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Importance of Pattern Recognition

The bearing defect frequency pattern is really the key in determining the defect severity for each bearing failure mode. The number of bearing-related harmonic frequencies present, possible sidebands, and time waveform data can be a much more reliable method of projecting needed bearing repairs.

Viewing time waveform data in acceleration units can be very helpful when diagnosing bearing defects. Inner and outer race defects typically show repeatable impacting patterns and ring down as the rolling elements are passing over the race defect.

The acceleration amplitude in the time waveform is also a valuable tool to use for diagnosing bearing severity. The acceleration amplitude is directly proportional to the forces affecting the bearing vibration. Remember, F = M * A. However, also remember that the machine's mass affects the amplitude variations from machine to machine. Usually time waveform acceleration levels in excess of 2 G's, peak-to-peak is significant enough to warrant the analyst's attention.

Although vibration amplitude is helpful in determining defect severity, the frequency pattern in the spectral data is often more indicative of progressive bearing damage than changes in spectral amplitude alone. Now the discussion that follows is the classic failure progression that applies to about 80% of all bearings. You should be aware that other defect scenarios do exist. However, the basic idea that amplitude alone cannot always be used to correctly identify bearings ready for replacement is still true.

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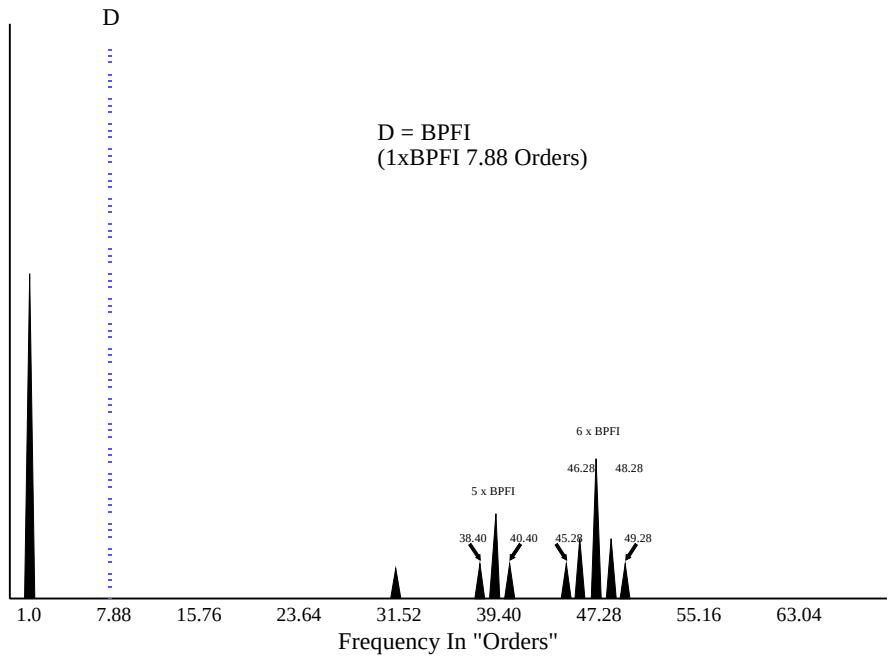
Failure Progression Pattern

The classical bearing failure progression pattern for most bearing race defects can be described as follows:

Mode 1

The first signs of a defective bearing will appear in the spectral data, typically at the higher frequency multiples of the fundamental bearing defect frequencies. One of the reasons that these high frequencies show up first is that the impacting of the rolling elements through the defect site creates high-frequency broadband noise that excites the high-frequency natural bearing resonances. The fundamental frequencies do not tend to show up at this point.



High frequency multiples will typically appear first with a race defect.

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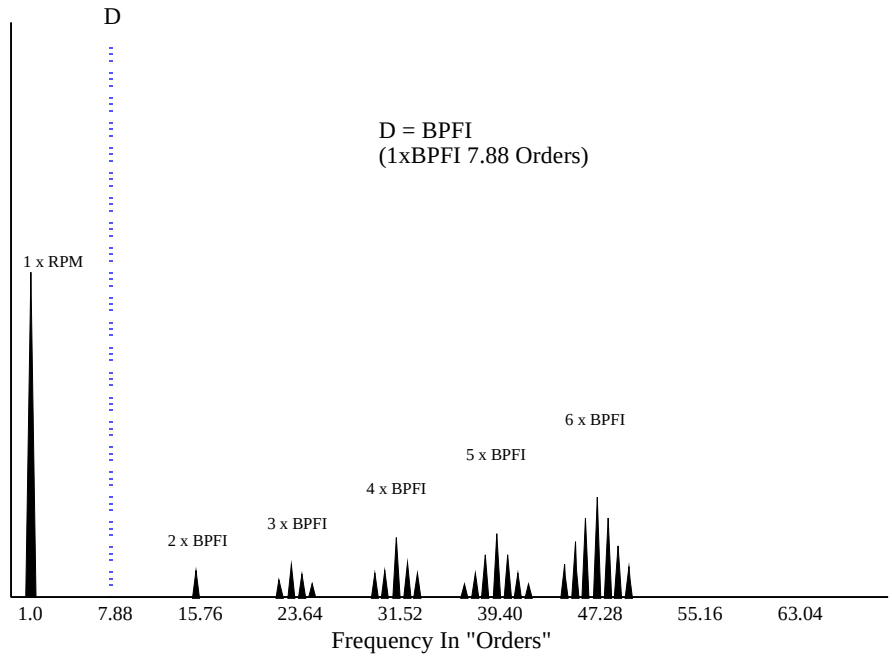
Mode 2

More harmonics of the defect frequency appear in the spectral data. Continuing degradation often means modulation of the defect frequencies by the shaft turning speed. The amplitude of the sidebands relative to the defect frequency harmonics is very important at this point. Sideband amplitudes exceeding the defect frequency harmonic amplitude can indicate significant damage. Please note, that as the bearings degrade, the actual defect frequencies are not going to exactly match the precalculated defect frequencies. This is because degradation of the bearing has caused the internal bearing geometry to deviate from the original design.



Defect progression includes multiple sidebands.

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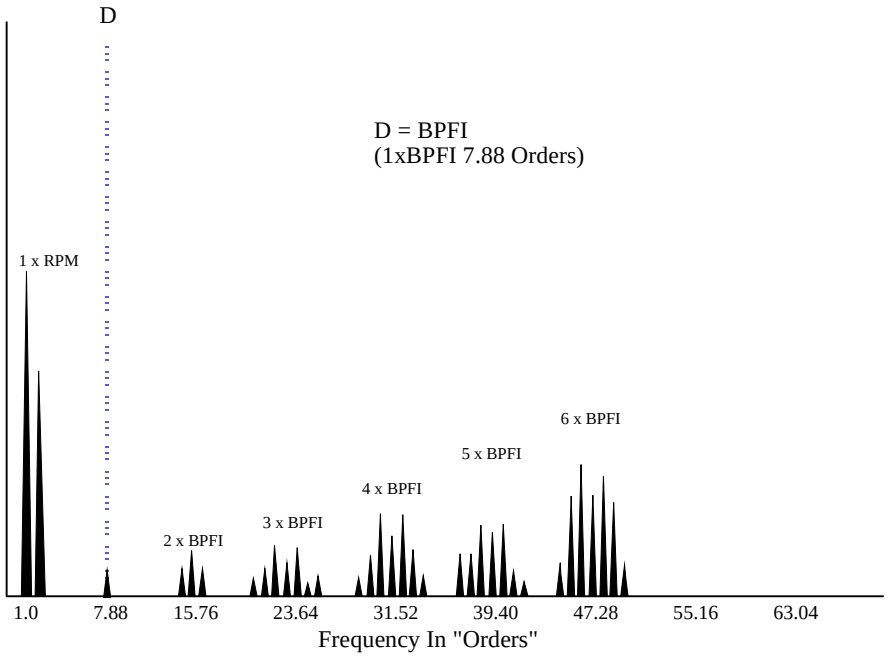
Mode 3

Mode 3 can be identified by the appearance of the fundamental bearing defect frequency in the spectral data in addition to the defect harmonics and sideband frequencies. Also, sideband frequencies may be present related to the BSF and the FTF frequencies. The BSF and the FTF usually appear as sidebands of the BPFI or the BPFO. This is found in the later stages of bearing degradation as the rolling elements and the train have become damaged from constant contact and contamination by material breaking loose from the defective raceways. The remaining bearing life has certainly been limited at this point, depending on the speed and the load.



The fundamental defect frequency now appears.

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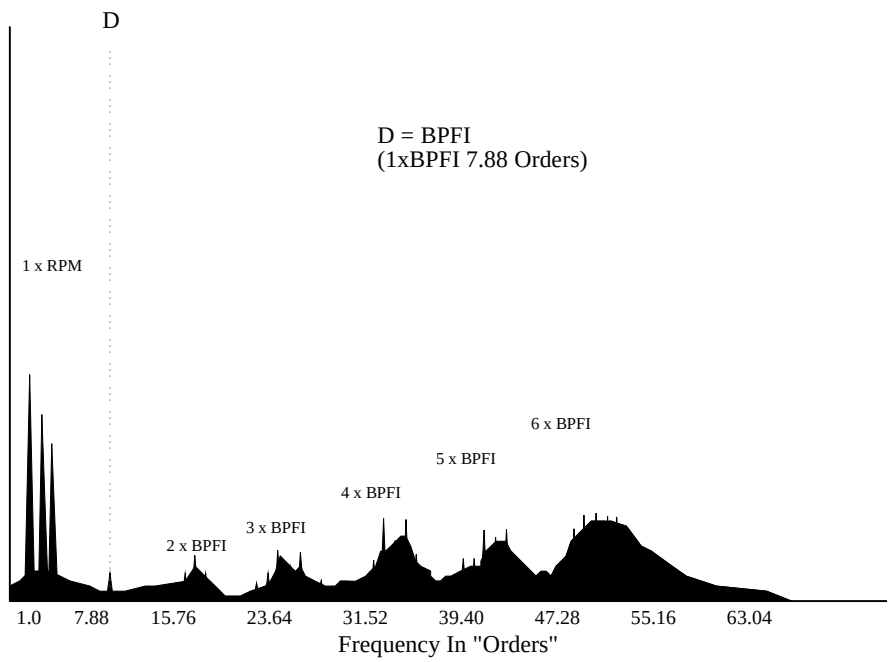
Mode 4

As the bearing continues to degrade, the internal clearances may begin to increase, which accelerates the deterioration of the bearing components. This allows more impacting to occur within the bearing, increasing the broadband noise. The increased impacting appears in the spectral data as a raised noise floor. The peaks in the data may seem to decrease in amplitude and become less distinct as the noise floor increases.



The noise floor has increased, making the peaks appear to have decreased. Failure is eminent at this point. Less than 2 percent of the bearing's life remains. If a crack develops in the inner or outer race, the internal clearances may decrease. This could lead to a seizure of the bearing. Steps to schedule the bearing's replacement should have been taken by this time. These steps may include ordering a new bearing and scheduling a suitable repair time.

It is important to understand that each bearing may progress through the failure modes at different rates. When a bearing defect mode is detected, data should be collected more often so all the modes are captured in the database. Every bearing failure is somewhat different. Often an analyst fails to capture some of the failure modes because they occur between data collection intervals.

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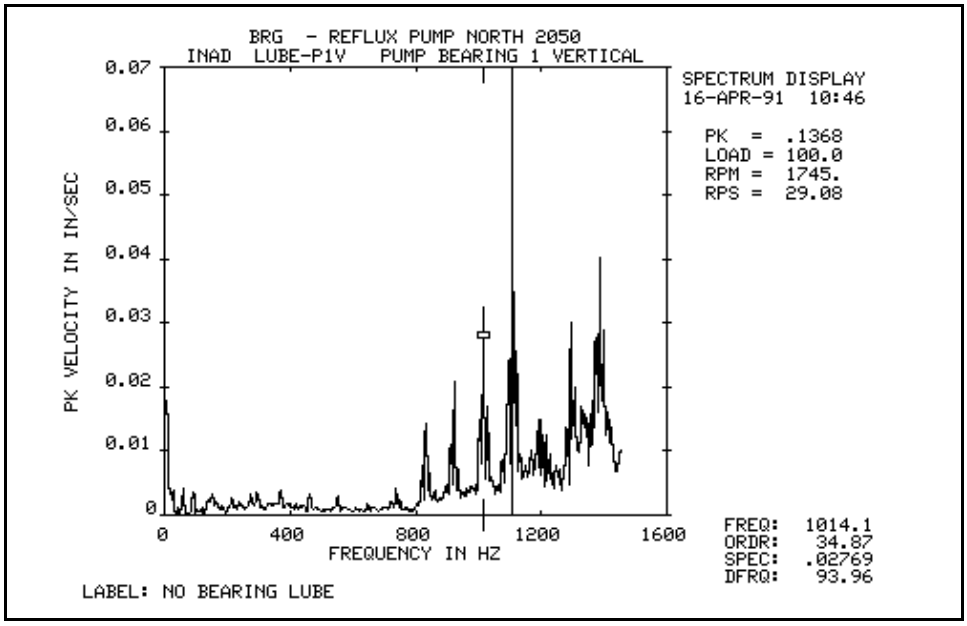
Lubrication Problems

A significant contributor to a defective bearing is improper lubrication. These defects often excite the bearing natural frequencies. The defect frequencies appear between 800 and 1200 Hz spaced 75 to 90 Hz apart as shown in the spectrum below.



Summary

Relying on amplitude alone is a difficult way to predict bearing failure. Often the bearing defect frequencies do not increase in amplitude as much as the content of the spectrum changes, including more of the defect frequencies.

Provided data is collected correctly to a suitable maximum frequency range, observing the failure progression pattern is a much more reliable method for recommending bearing replacement than trying to rely on amplitude alone. Using the analysis techniques discussed can provide a vibration analyst with a better guide for determining when to replace defective bearings.

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Bearing Defect Chart

(page one of three)

Frequency	Defect	Remarks
1-8 x BPFI	inner race defect	amplitude of harmonics often exceeds that of fundamental frequency; defects frequently caused by forces from the rotating element; 1xTS sidebands often modulate it as bearing degrades
1-8 x BPFO	outer race defect	amplitude of harmonics often exceeds that of fundamental frequency
1-N x BSF	ball or roller defects; sometimes result from a broken cage (N = number of rolling elements)	usually accompanied by race defects; the strongest multiple frequently equals the number of defective rolling elements

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Bearing Defect Chart

(page two of three)

Frequency	Defect	Remarks
FTF	cage defect	usually accompanied by other defective components; may appear as a difference frequency
1xTS modulation (sidebands) or significant broadband energy	advance defects resulting in alterations of bearing geometry	energy centers around initiating defect frequency, although this frequency may disappear in advanced degradation
sum and difference frequencies (sidebands) involving RPM, BPFI, BPFO, BSF, FTF	multiple defective parts	extensive damage

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Bearing Defect Chart

(page three of three)

Frequency	Defect	Remarks
1-6xTS	(1) excessive internal clearances	excessive clearances normally accompanied by FTF modulating other frequencies; can also significantly affect balance sensitivity
	(2) bearing turning on shaft	frequently 3xTS or greater is the predominant multiple
	(3) bearing loose in housing	shows strong 1xTS and 4xTS
	(4) misaligned bearing	generated frequency equals number of rolling elements xTS
900-1600 Hz region with 3-4 peaks separated by **80-130Hz**	inadequate lubrication	amplitudes may grow to 0.1-0.2 IPS; may result from excitement of installed natural frequencies; can also be caused by preloading or heavy thrust load if lubrication does not correct

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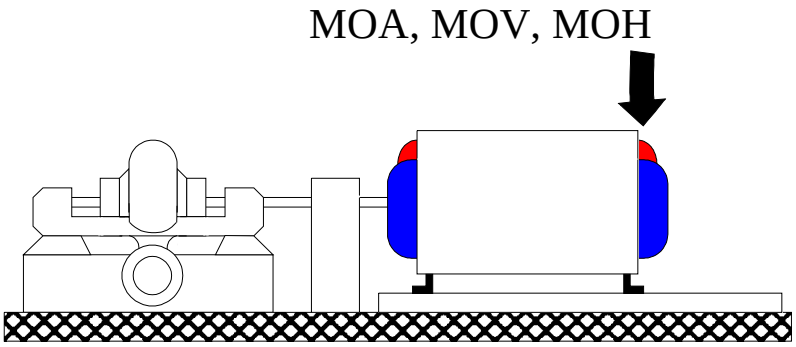
Case Histories

Bearing Case #1



❶ 250 HP, 1200 RPM motor.

❷ Center Hung - double suction pump.

❸ Most of the energy in the axial direction.

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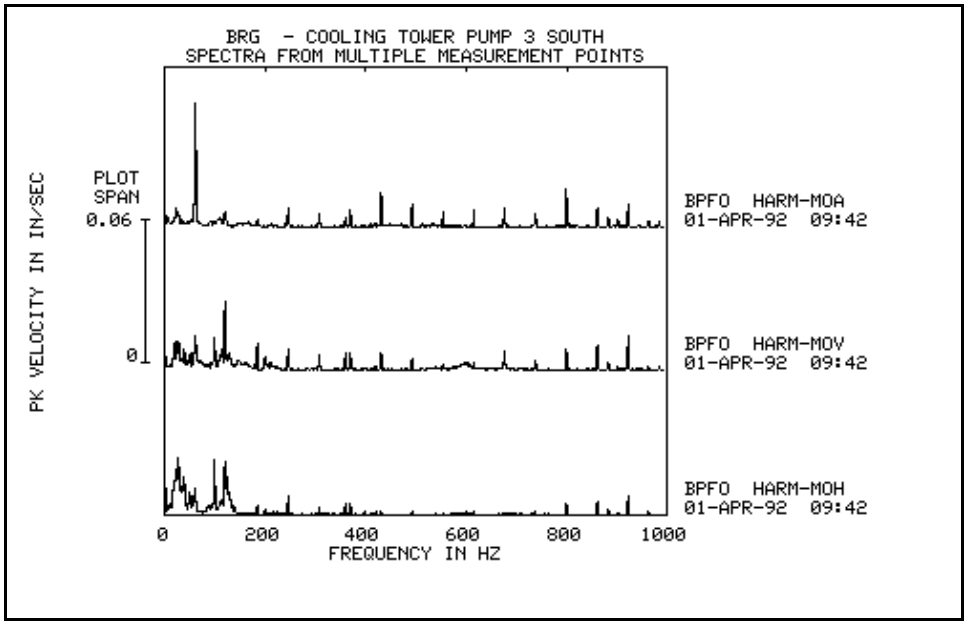
Bearing Case #1



The multiple-point plot shows the outboard points on the motor. This is the first time data was collected on this machine. The highest amplitudes are seen on the Motor Outboard Axial (MOA) point. Notice the appearance of the higher frequency peaks.

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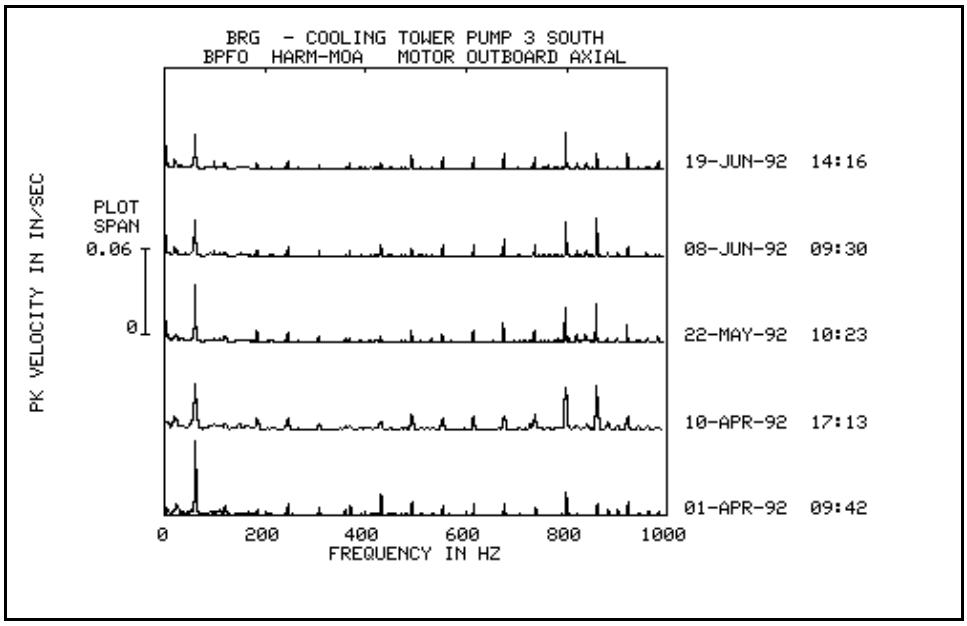
Bearing Case #1



The above plot shows the data from April to June on the MOA point. Notice the amplitudes of the higher frequency peaks are increasing as the lower frequency peaks are decreasing in amplitude. The next page shows a full screen of the data taken in June.

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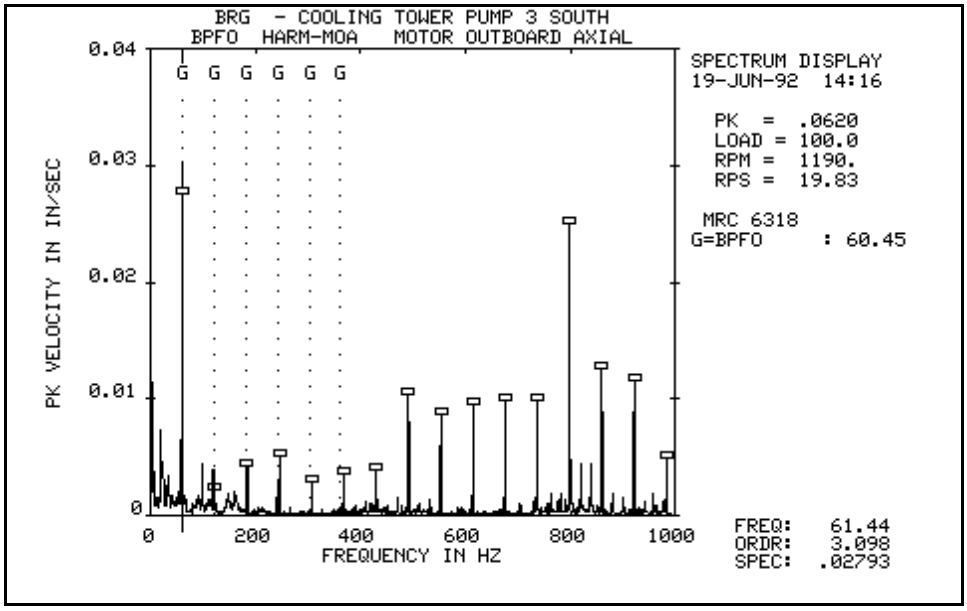
Bearing Case #1



The above plot shows a full screen spectrum of the MOA point. The fault frequency for the BPFO is displayed. Notice the primary BPFO is present in the above spectrum. Typically, primary BPFO is not seen until the final stages of the bearing defect, if it is seen at all. What should the waveform look like?

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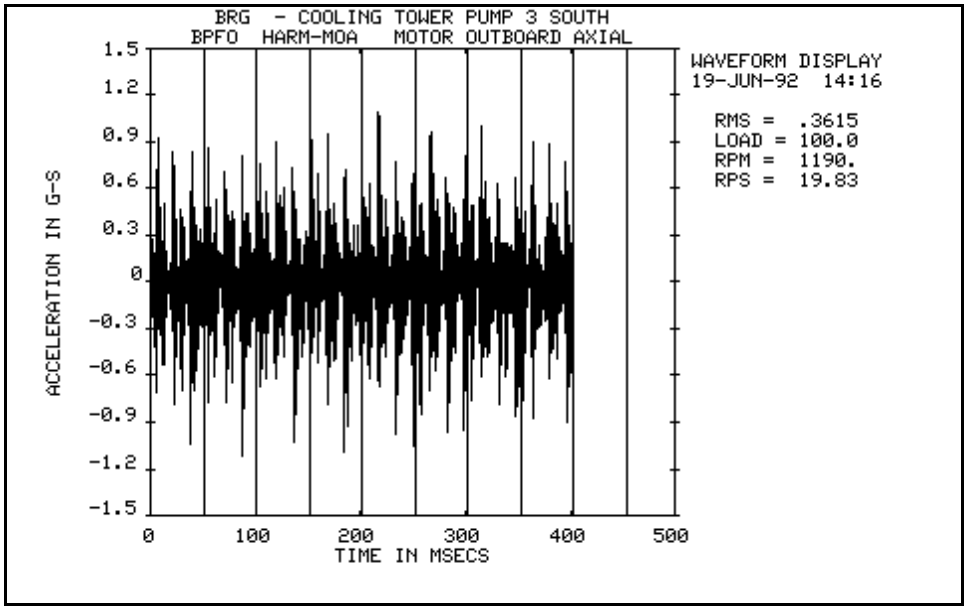
Bearing Case #1



The above plot shows the waveform from the MOA point. Notice the typical waveform pattern of impacting and ringing down. However, the amplitude is only about 0.9 g's. This case illustrates why severity determination cannot be based solely on amplitude levels.

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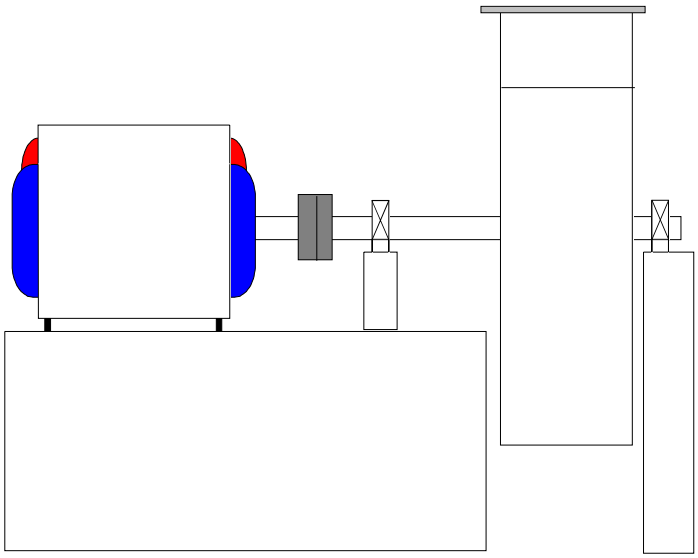
Bearing Case #2



❶ 250 HP, 3600 RPM motor.

❷ These readings were taken for certification.

❸ Spectral data now shows a new problem.

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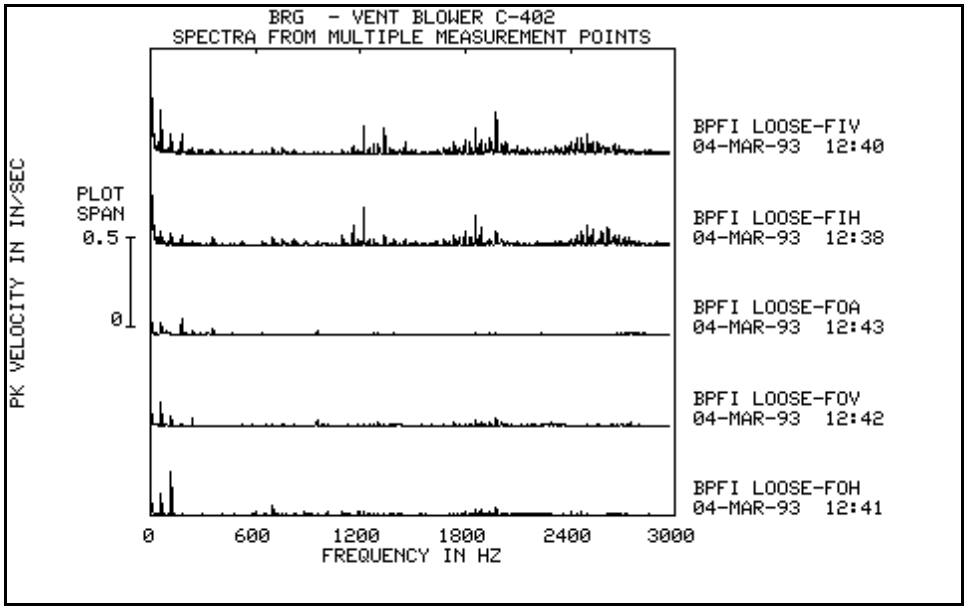
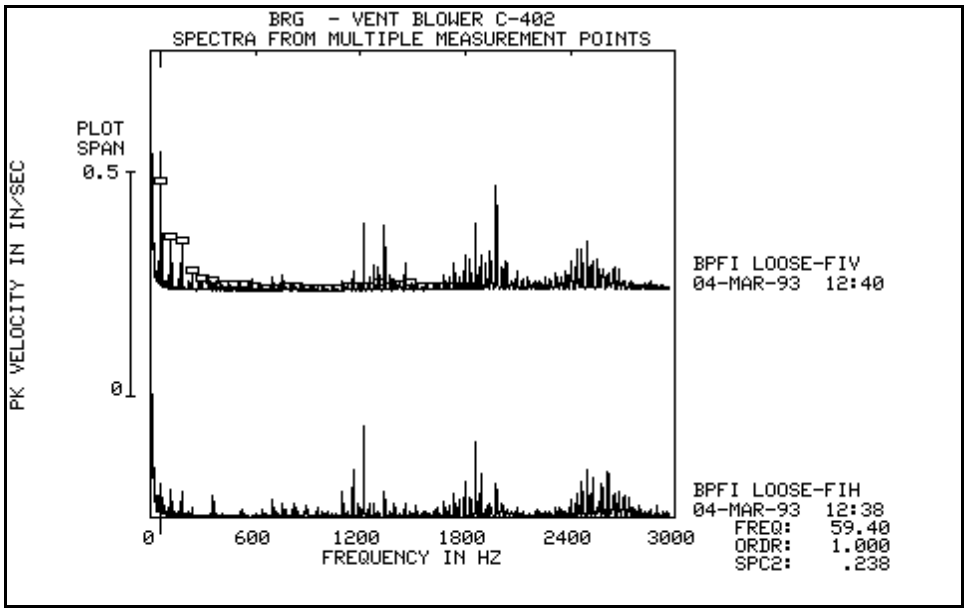
Bearing Case #2



The above plot shows a multiple spectra plot of all of the Fan points. Notice the higher frequency activity on the inboard points.

Bearing Case #2



The above plot shows only the inboard points. Notice the harmonics of 1xTS and the high frequency nonsynchronous groups of peaks.

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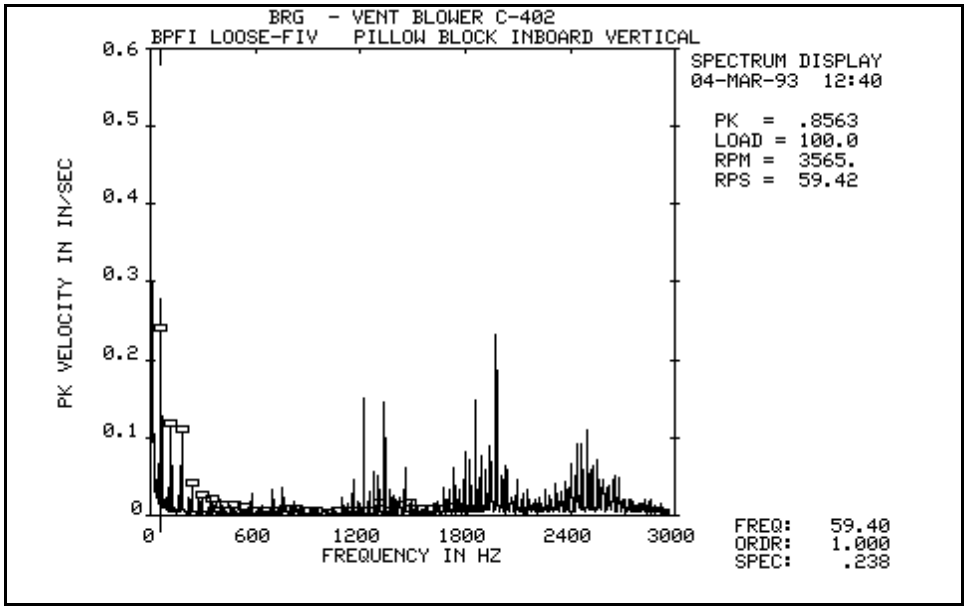
Bearing Case #2



The plot above shows a single spectrum of the FIV point. The bearing ID is unknown, however, characteristics in the spectrum indicate an inner race defect. Based on the spectral pattern, the waveform should be examined to confirm the suspected inner race defect.

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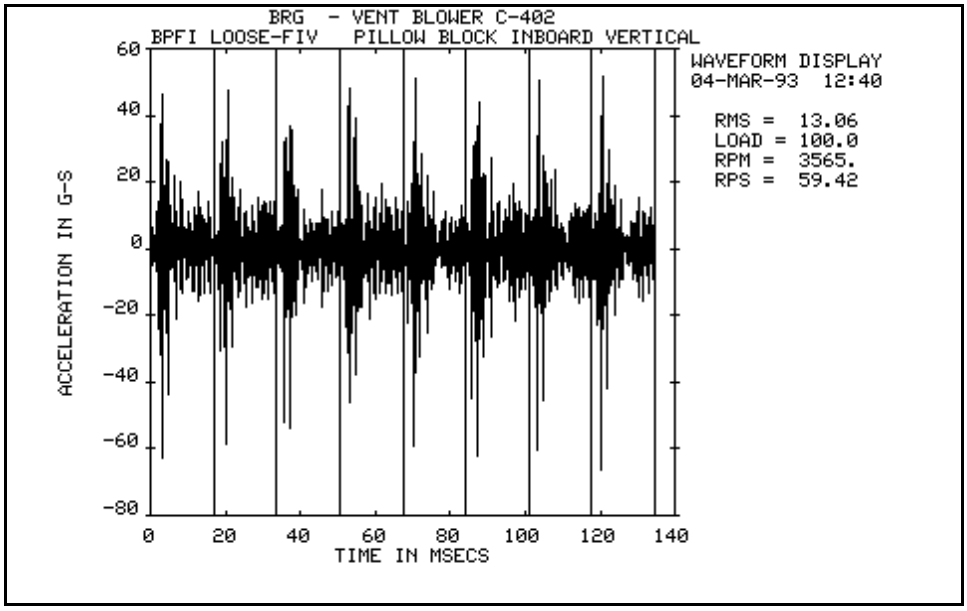
Bearing Case #2



The waveform for the FIV point shows significant impacting of ± 60 g's. Production required the unit to run through the weekend before any work could be performed. The bearing failed within 24 hours of data collection.

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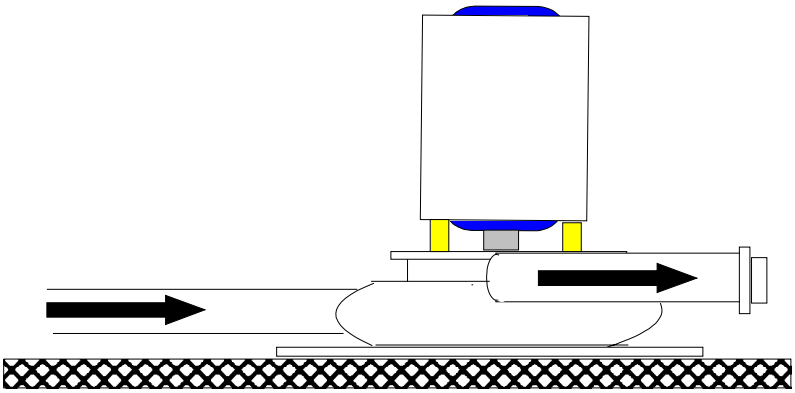
Bearing Case #3



❶ 40 HP, 1800 RPM motor.

❷ Vibration increased from last month.

❸ Vertical mounted centrifugal pump with 7 blades.

❹ Motor is attached to pump by a spline connection.

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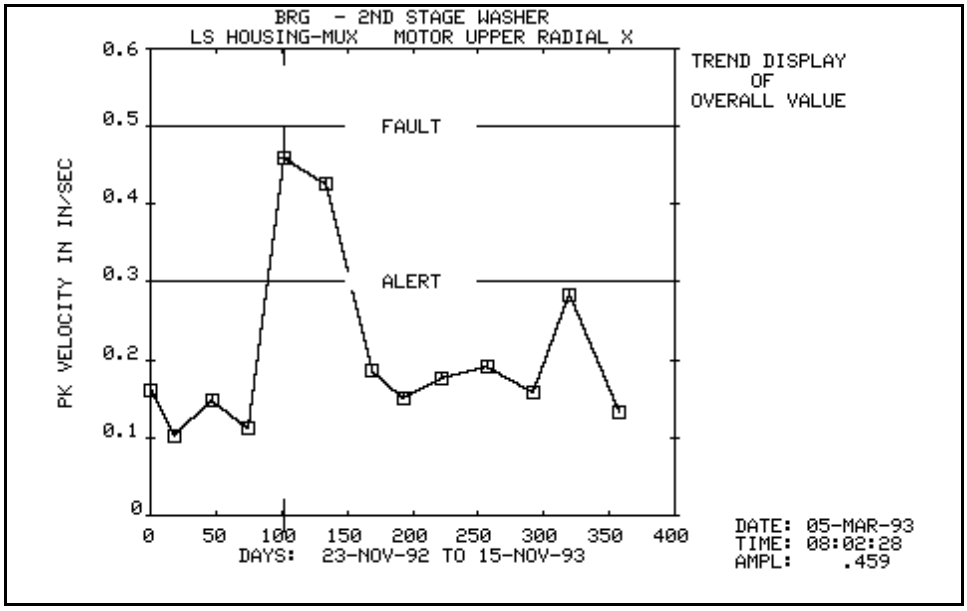
Bearing Case #3



The above plot shows a trend plot of the overall parameter value for the Motor Upper radial X (MUX) measurement point. The cursor is located at the March 5, 1993, point. The overall trend plot shows that the overall value increased on March 5. A multiple parameter plot is needed to determine where the increase occurred.

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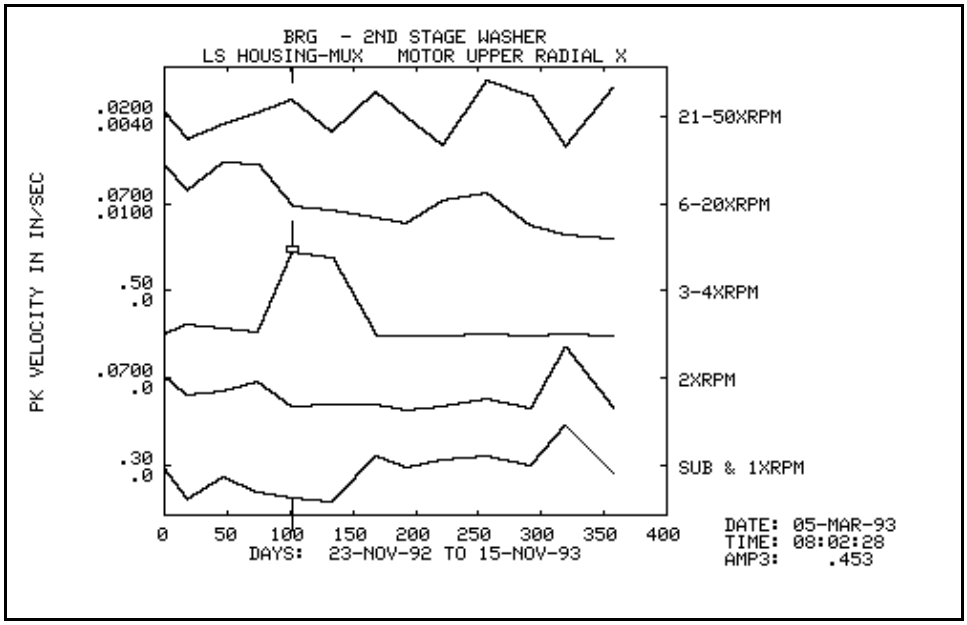
Bearing Case #3



The above plot shows the multiple parameter plot for the MUX point. According to the plot, the parameter that caused the sudden increase in overall value is the 3 to 4xTS parameter band.

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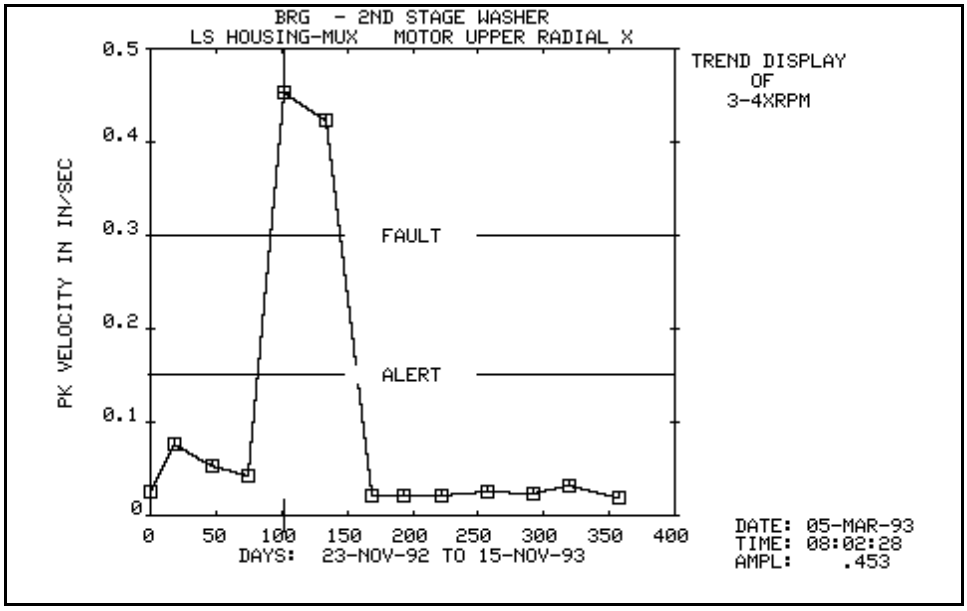
Bearing Case #3



The trend plot above shows the 3 to 4xTS parameter band. This sudden increase is cause for concern. The spectrum and waveform need to be analyzed to determine the cause.

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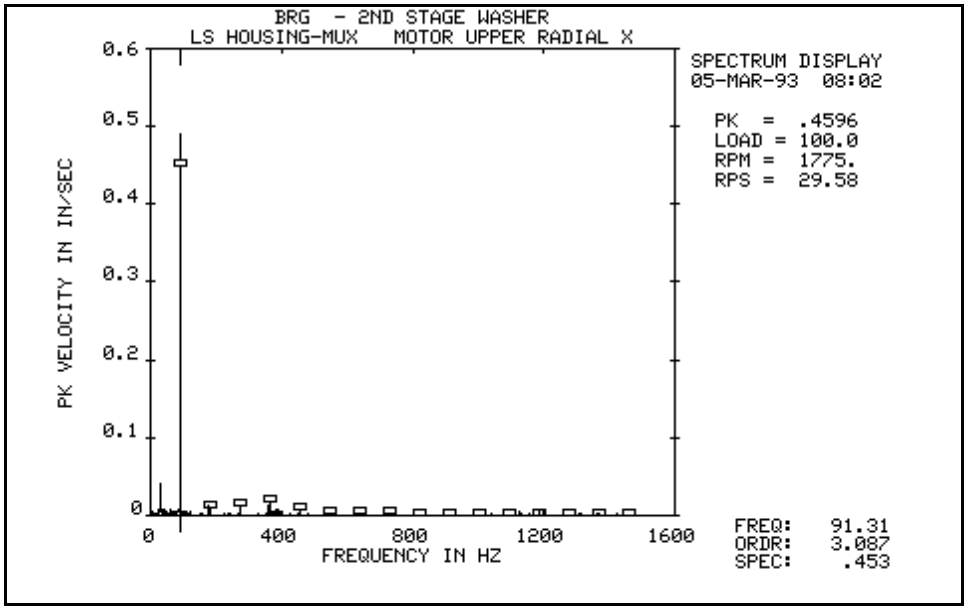
Bearing Case #3



The single spectrum above shows most of the energy in the spectrum is located at 3xTS. The source of the 3xTS needs to be determined. Based on information known (7 blades, no coupling), the analyst determined the 3xTS *must* be a result of bearing housing looseness. He was determined to find out what was causing this peak at 3.097 Hz.

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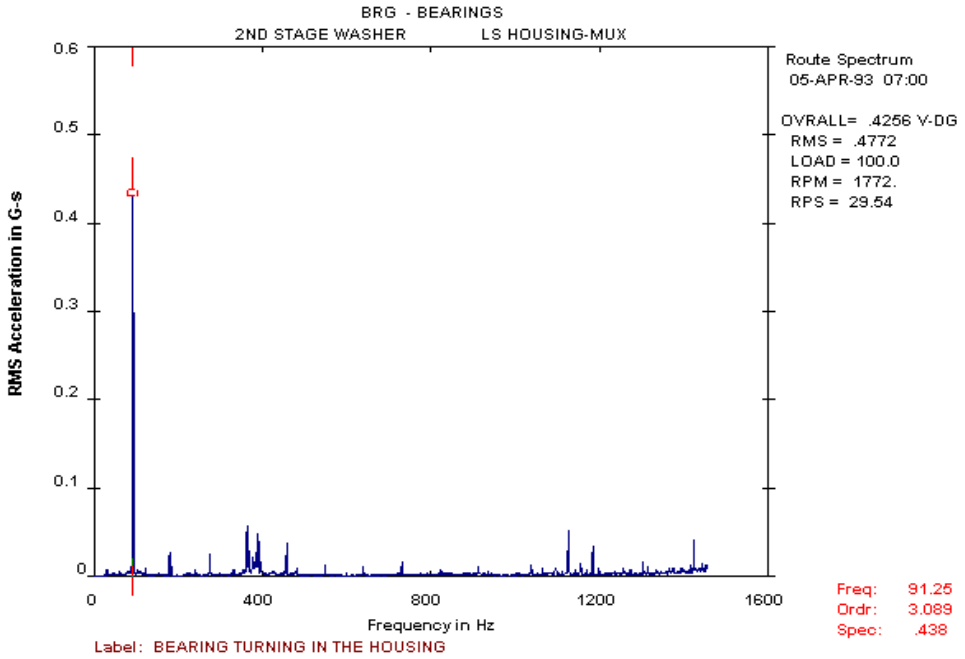
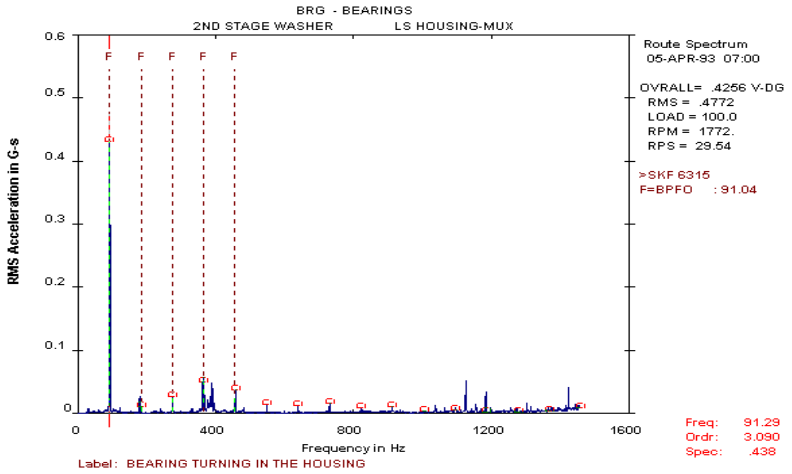
This is a good example of using acceleration to help solve a possible bearing problem. The units are now displayed in acceleration. Acceleration reveals peaks at higher frequencies. What are these peaks?



Fault frequencies can sometimes be indispensable. The spectrum below displays the fault frequencies for the BPFO.



The Analyst also used the peak list to help find the problem.

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LIST OF SPECTRAL PEAKS
****

Machine: (BRG) 2ND STAGE WASHER
Meas. Point: LS HOUSING-MUX --> MOTOR UPPER RADIAL X
Date/Time: 05-APR-93 07:00:26 RPM= 1772. Units=G-s RMS
Data Label: BEARING TURNING IN THE HOUSING

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	91.25	.4384	3.09	13	729.97	.0162	24.71
2	177.83	.0295	6.02	14	1038.64	.0148	35.16
3	273.75	.0310	9.27	15	1126.34	.0594	38.13
4	359.85	.0536	12.18	16	1154.78	.0175	39.09
5	364.99	.0645	12.36	17	1184.13	.0354	40.08
6	376.34	.0245	12.74	18	1304.03	.0182	44.14
7	379.76	.0203	12.86	19	1314.66	.0139	44.50
8	385.26	.0351	13.04	20	1415.91	.0142	47.93
9	389.50	.0506	13.18	21	1424.04	.0433	48.21
10	394.46	.0203	13.35	22	1445.57	.0168	48.93
11	456.26	.0390	15.44	23	1453.68	.0159	49.21
12	547.52	.0136	18.53	24	1457.20	.0142	49.33

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
.4653	0.0 / 0%	.0209 / 0%	.4648 /100%
(ELECTRICAL=	0.00E+00/ 0%	FLOOR= .0758	SPEC CREST= 1.09)

Note the percentage of nonsynchronous energy in this peak list. What was the problem? This was an unusual case. When the machine was taken down for repairs the motor outboard bearing did have BPFO's and the bearing was also loose in the housing. This would explain why the peak at 3.087 Hz was at such a high amplitude (which was the fundamental frequency for the BPFO. Unusual because of the amplitude of the fundamental BPFO compared to the harmonics of that frequency.

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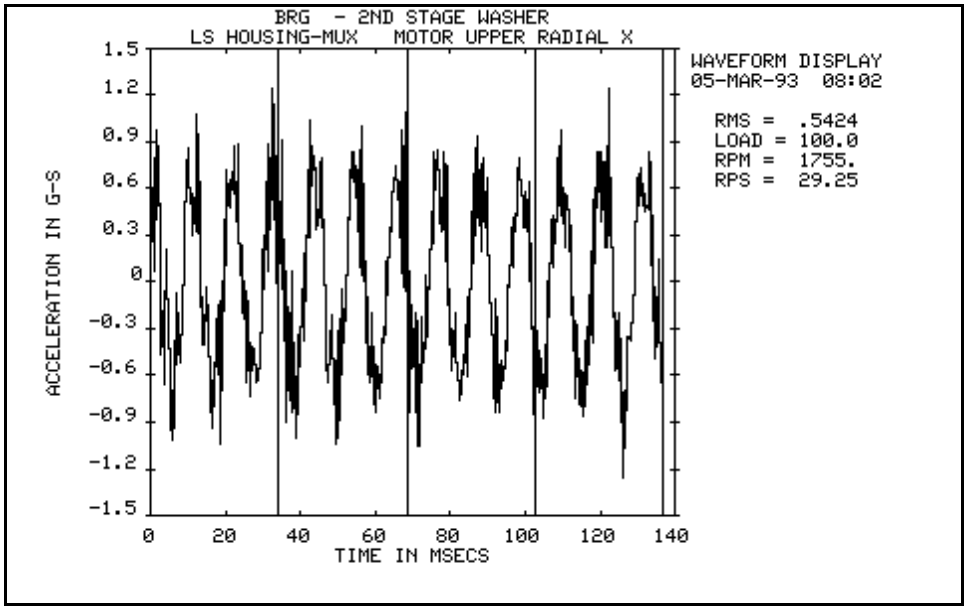
Bearing Case #3



The waveform shows the sinusoidal pattern with three distinct peaks per shaft revolution. This repetitive shape, however, lacks complexity and the amplitudes of peaks remain relatively low.

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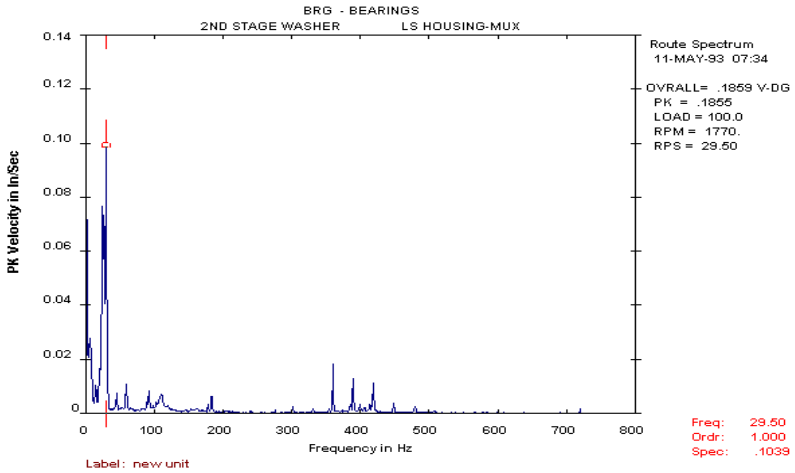
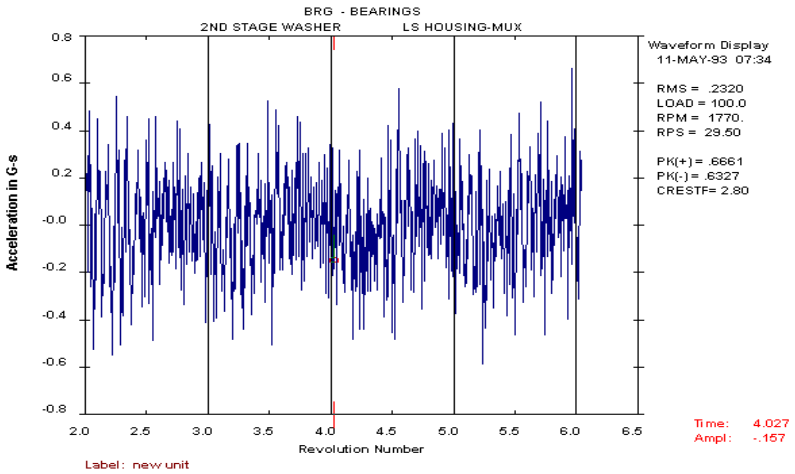
This is a display of the spectrum after repairs were made. The vibration has been reduced by more than ½ .



We have a completely different type waveform pattern also.



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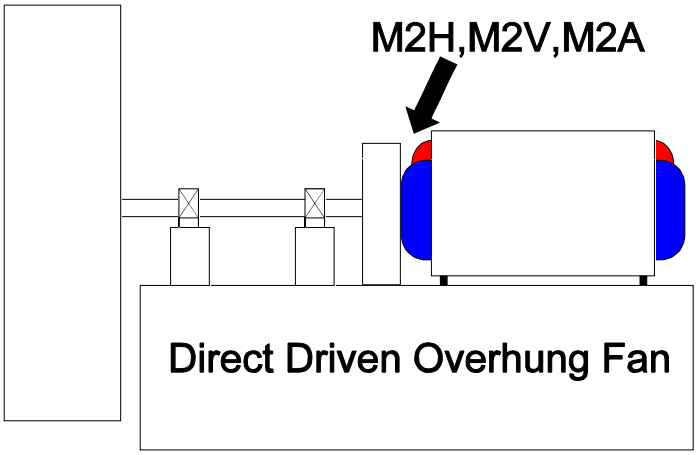
Bearing Case #4



❶ 200 HP, 1200 RPM motor.

❷ Overhung fan - direct driven.

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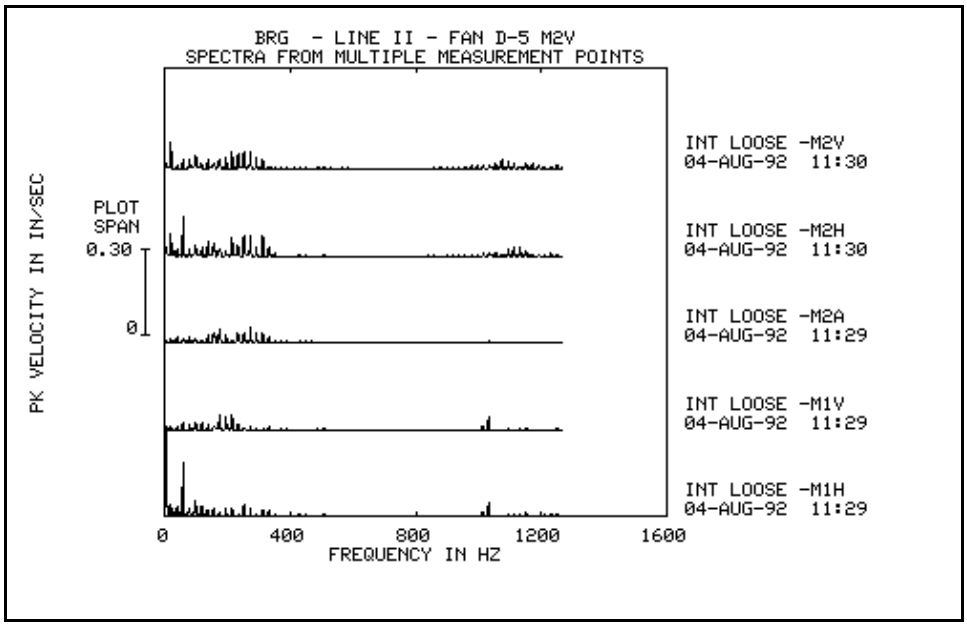
Bearing Case #4



The above multiple-point plot shows all the motor points. Most of the activity is occurring at the inboard points.

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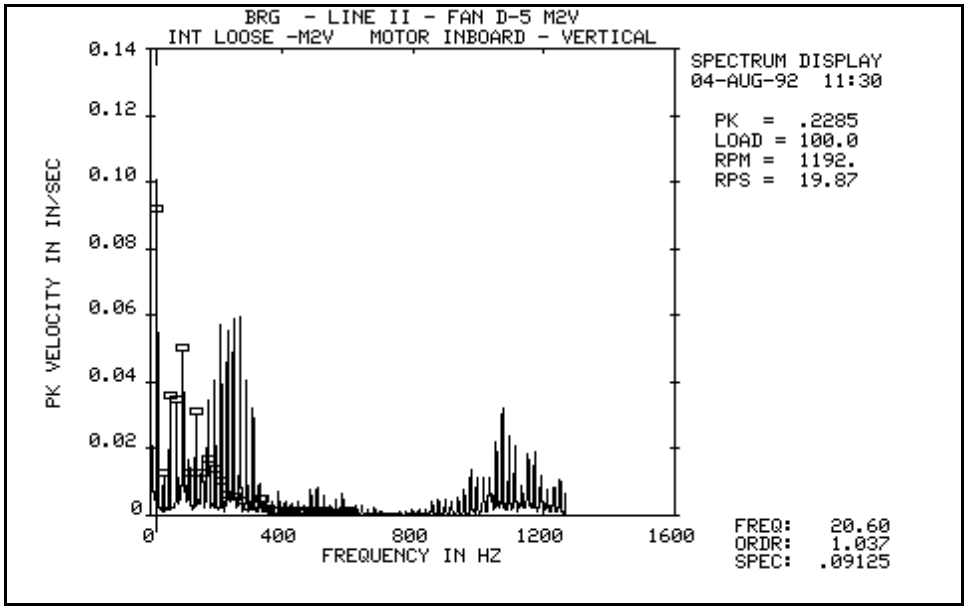
Bearing Case #4



The above single spectrum is shown for point Motor Inboard Vertical (M2V). Several multiples of turning speed are identified by the harmonic cursor. This is an indication of looseness.

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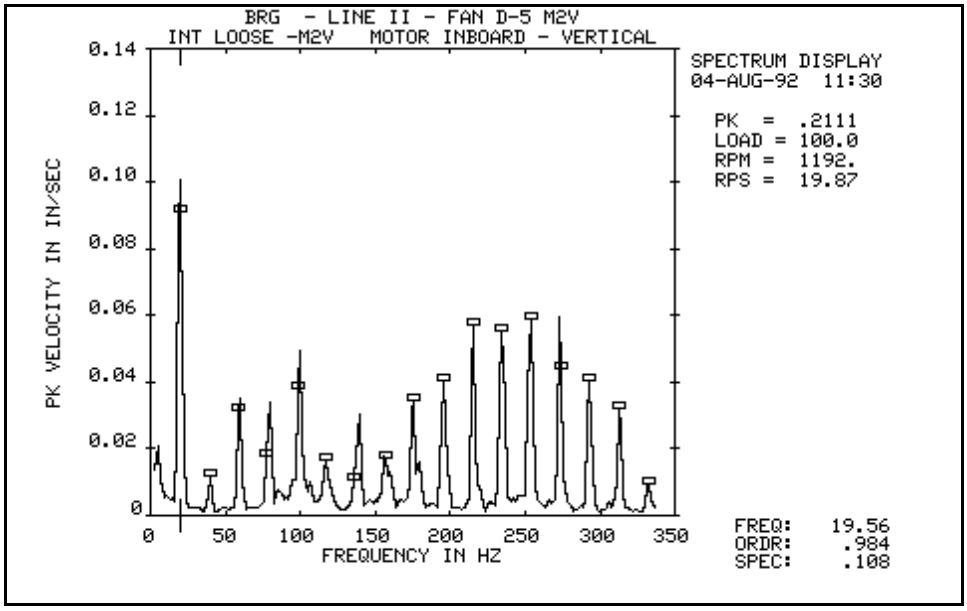
Bearing Case #4



The above spectrum is an expanded plot of the previous spectrum. Notice the pattern shows an increase in amplitude of the harmonic peaks as you move away from shaft turning speed.

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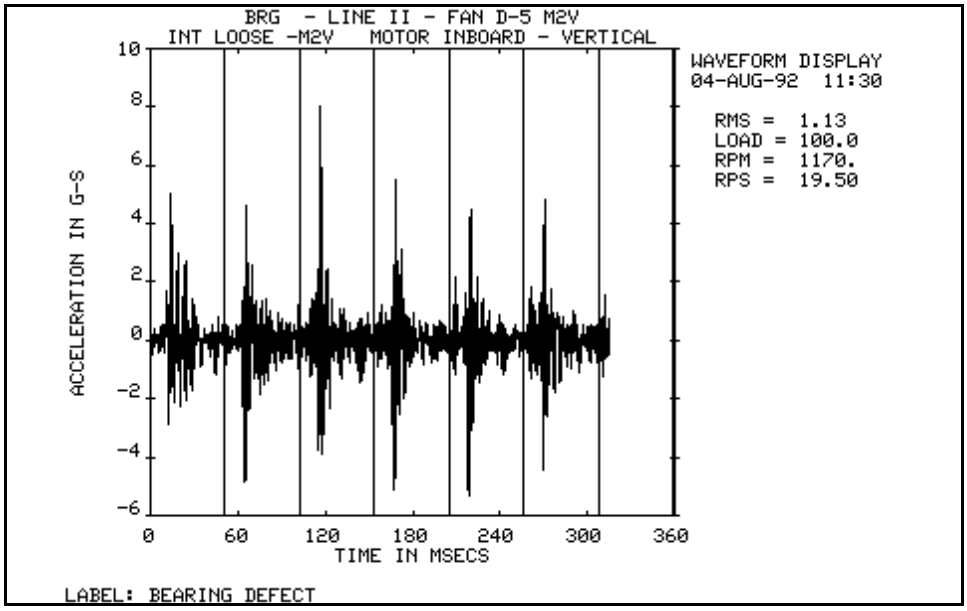
Bearing Case #4



The waveform shows the impacting and ringing down pattern. Also, notice the level of g's indicated in the waveform. Why is the waveform showing this pattern instead of a random looseness pattern? This is an indication of bearing looseness showing impacting in the waveform as opposed to structural looseness which typically shows more of a random pattern in the waveform.

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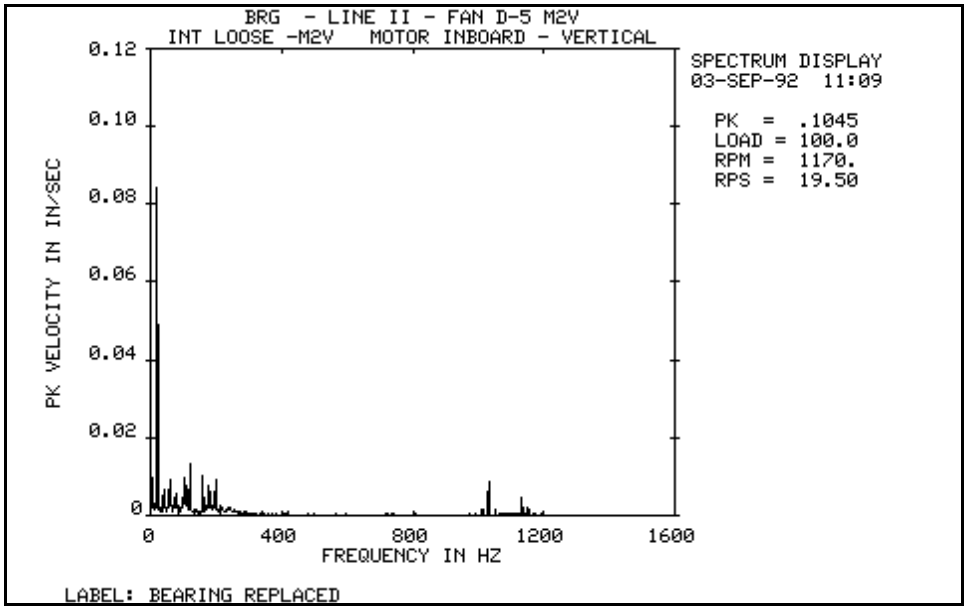
Bearing Case #4



The above spectrum was taken after the bearing was replaced. Notice the drop in overall energy as well as the decrease in harmonic activity.

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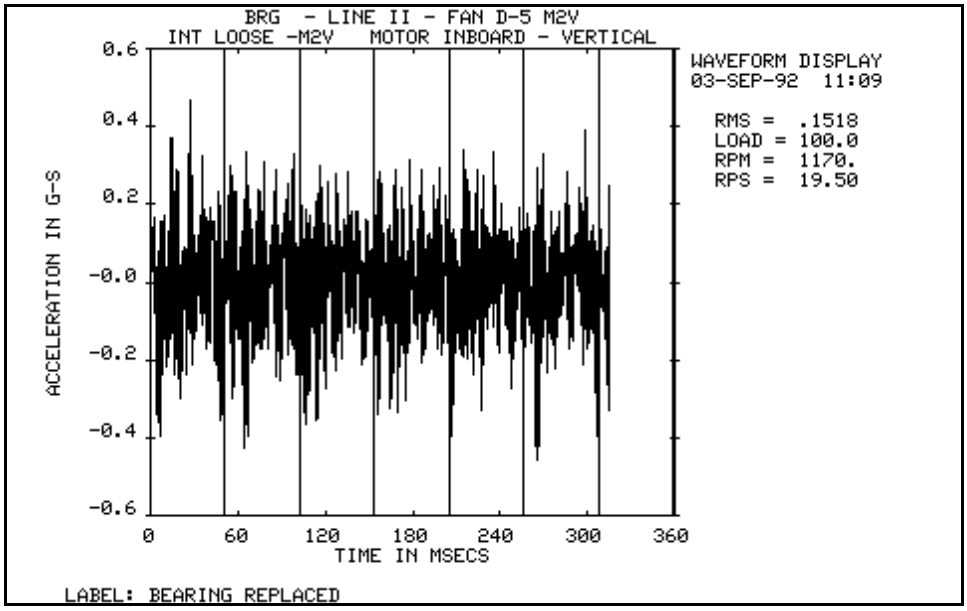
Bearing Case #4



The waveform is showing very little activity after the bearing was replaced. The impacting and ringing down pattern has disappeared.

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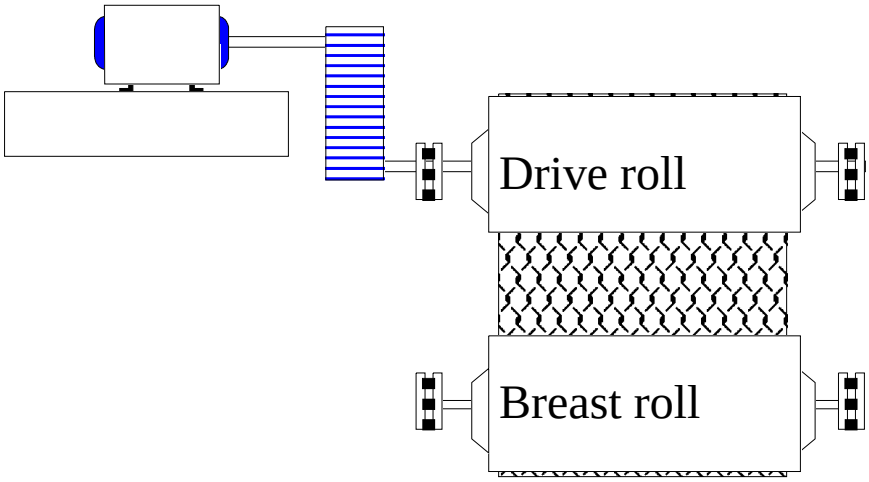
Bearing Case #5



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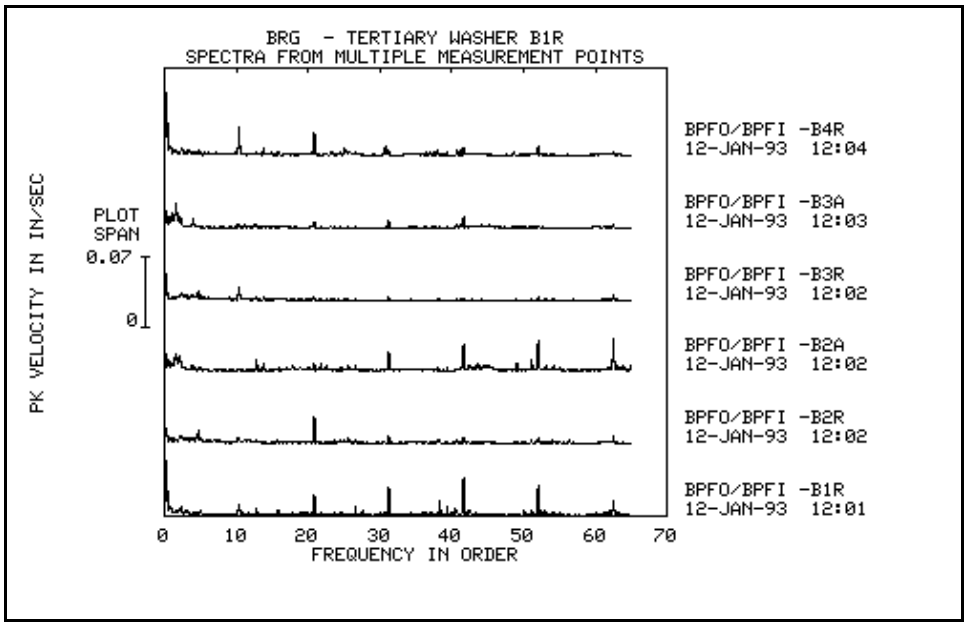
Bearing Case #5



The above plot shows a multiple measurement point plot. Notice the high frequency groups of peaks on point B1R.

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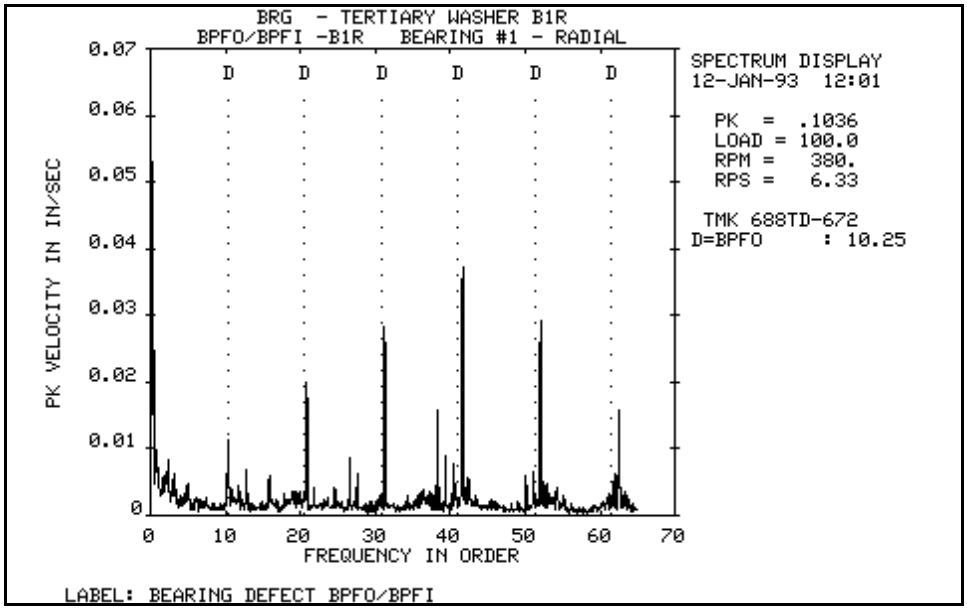
Bearing Case #5



The bearing frequencies were calculated as FTF = .45, BSF = 4.36, BPFO = 10.25, and BPFI = 12.75. The single spectrum shown above has the fault frequency for BPFO displayed. Six multiples of BPFO are present in the spectrum.

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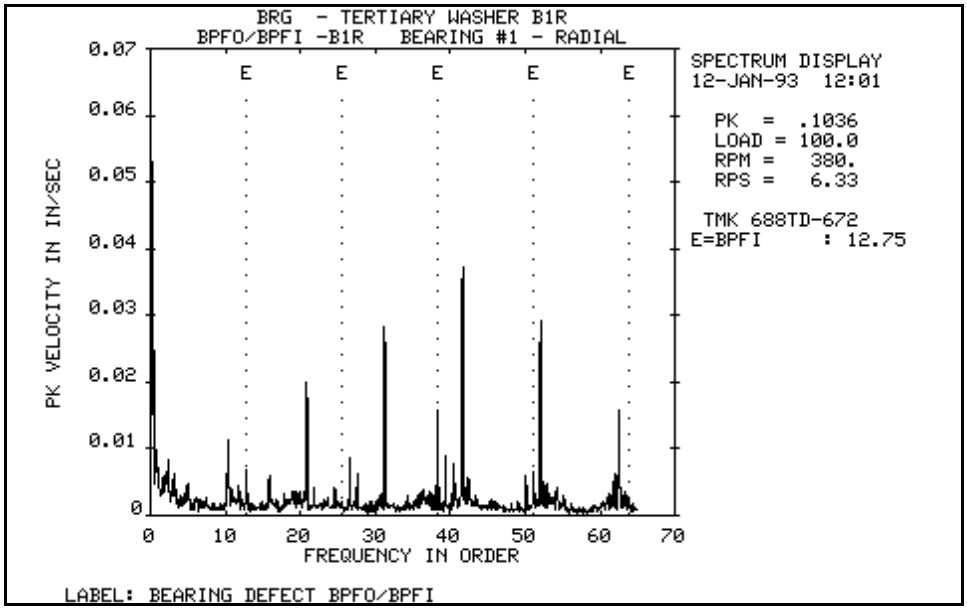
Bearing Case #5



The single spectrum shown above has the BPFI fault frequency displayed. Notice that both the BPFO and BPFI frequencies are present. What could the spacing of the sidebands be equal to?

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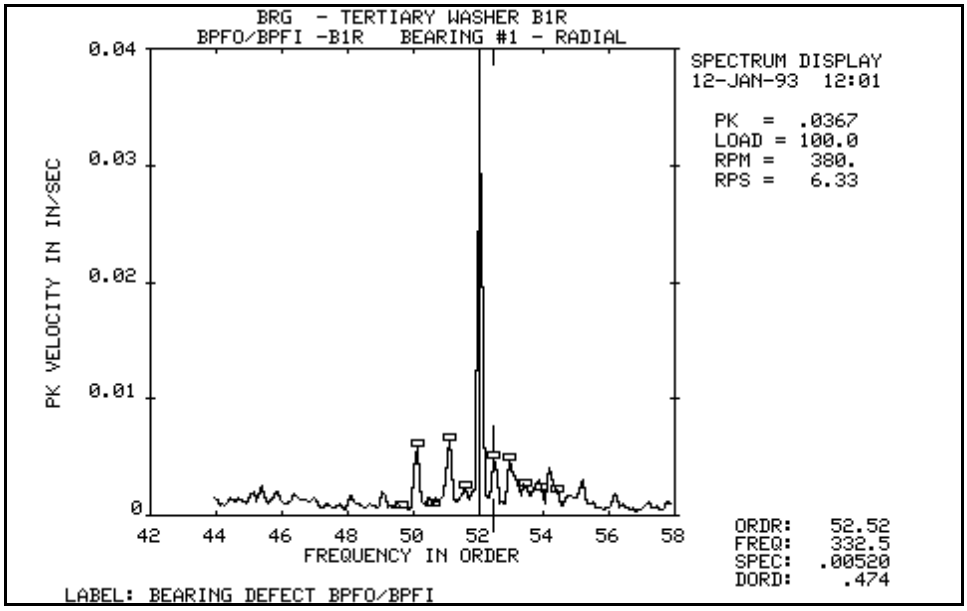
Bearing Case #5



The above spectrum is an expanded view of the spectrum from the previous page. The sidebands are spaced around BPFO at 0.47 orders. The sidebands are spaced at cage frequency. This is an indication of extensive bearing damage.

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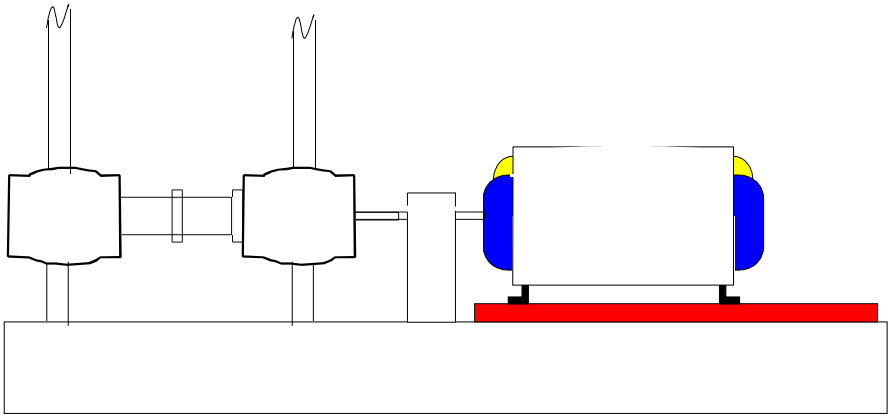
Bearing Case #6



❶ 40 HP, 1800 RPM motor.

❷ Motor drives 2 hydraulic pumps.

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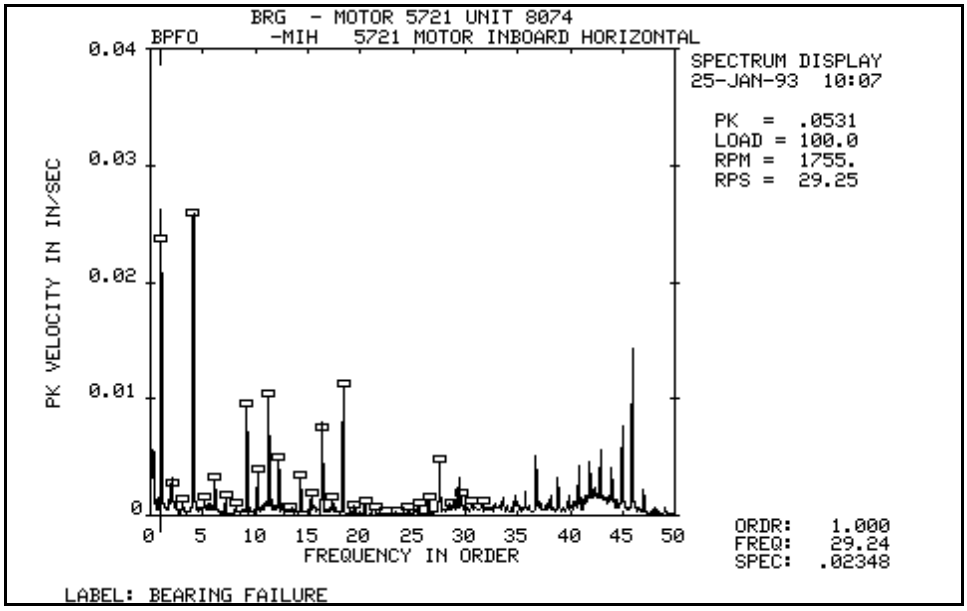
Bearing Case #6



The above spectrum shows several harmonics present of 1xTS. If this spectrum was not analyzed more closely, it could be misdiagnosed as a looseness problem. The next page uses fault frequency information to identify the real defect.

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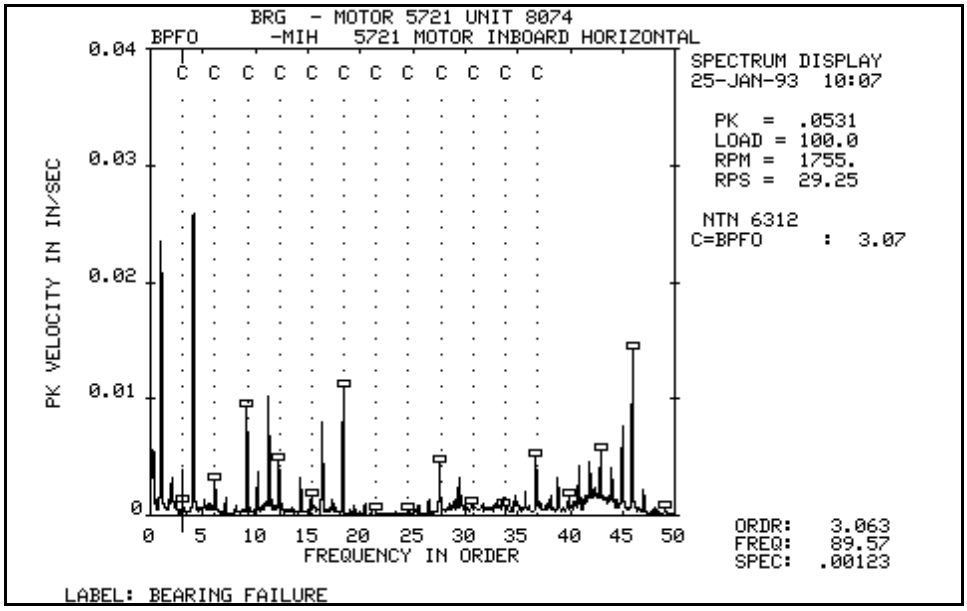
Bearing Case #6



The spectrum shown above has displayed the fault frequencies of BPFO on the spectrum. Also, harmonics of BPFO have been shown with the harmonic cursor markings. Notice that 16 harmonics of BPFO are present. This bearing defect could be overlooked because of the absence of the typical bearing defect pattern.

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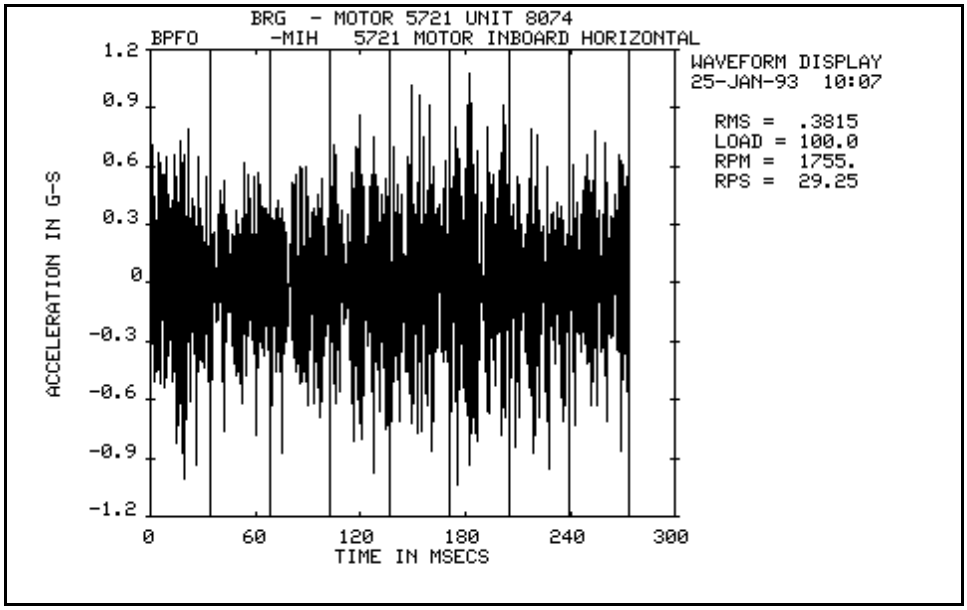
Bearing Case #6



Even the waveform does not show the typical pattern of bearing defect. This waveform has fairly low energy levels and a random nonrepetitive pattern. However, this waveform does show impacting of ± 1 G.

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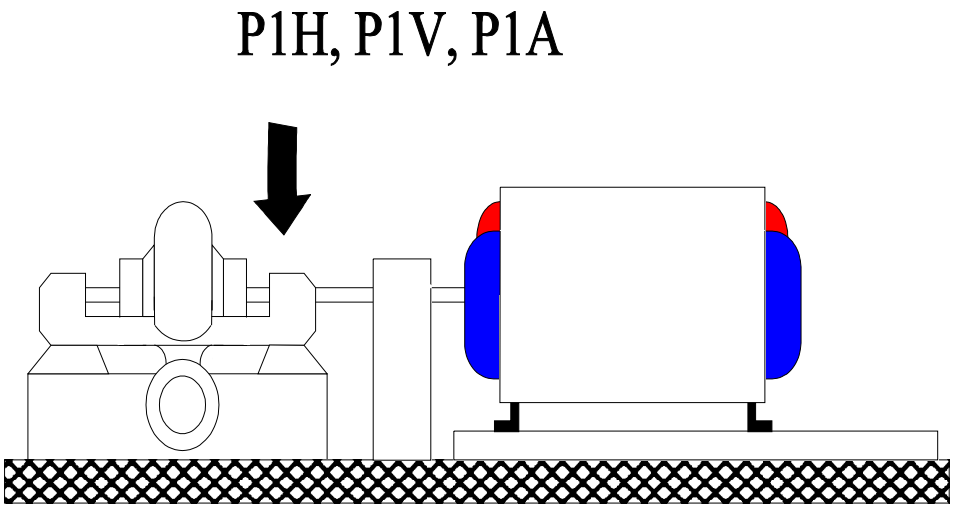
Bearing Case #7



❶ 125 HP, 1800 RPM motor.

❷ Center hung pump.

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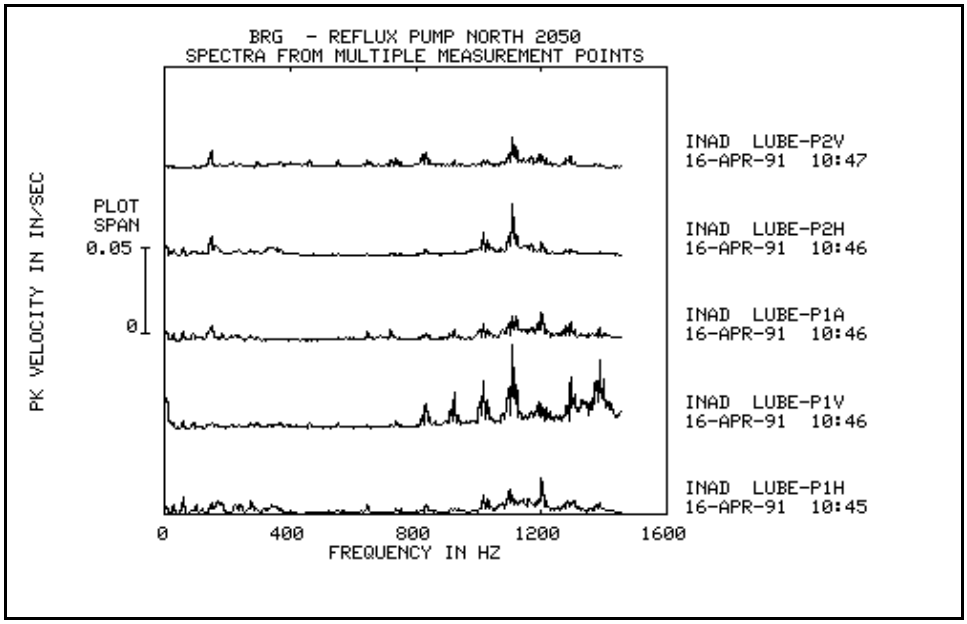
Bearing Case #7



The above multiple point plot shows all the pump points. Notice the high-frequency mounds of energy between 800 and 1200 Hz. A full screen of point P1V is shown on the following pages.

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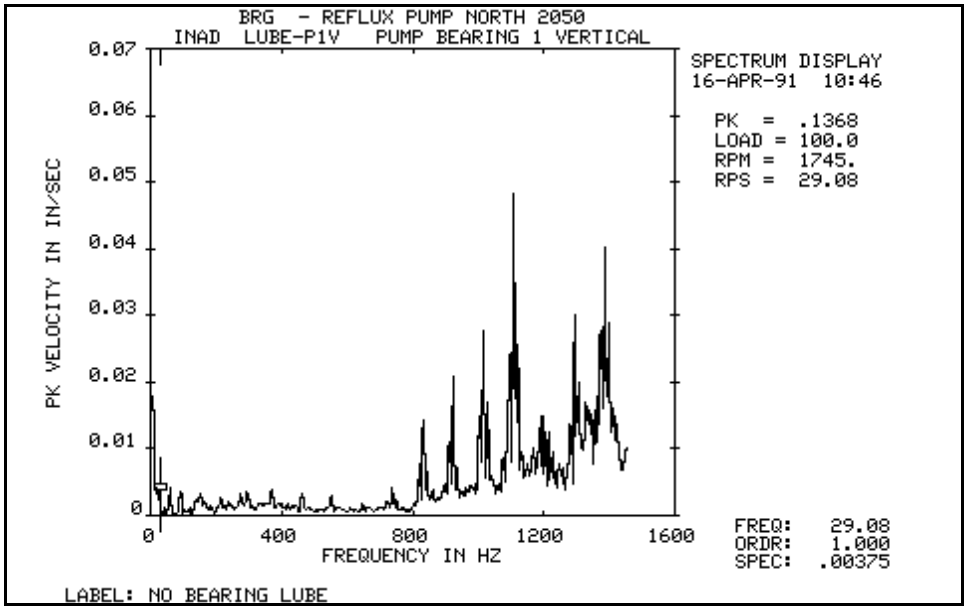
Bearing Case #7



The above full-screen spectrum shows a classic example of inadequate lubrication. Most of the energy is located between 800 and 1200 Hz.

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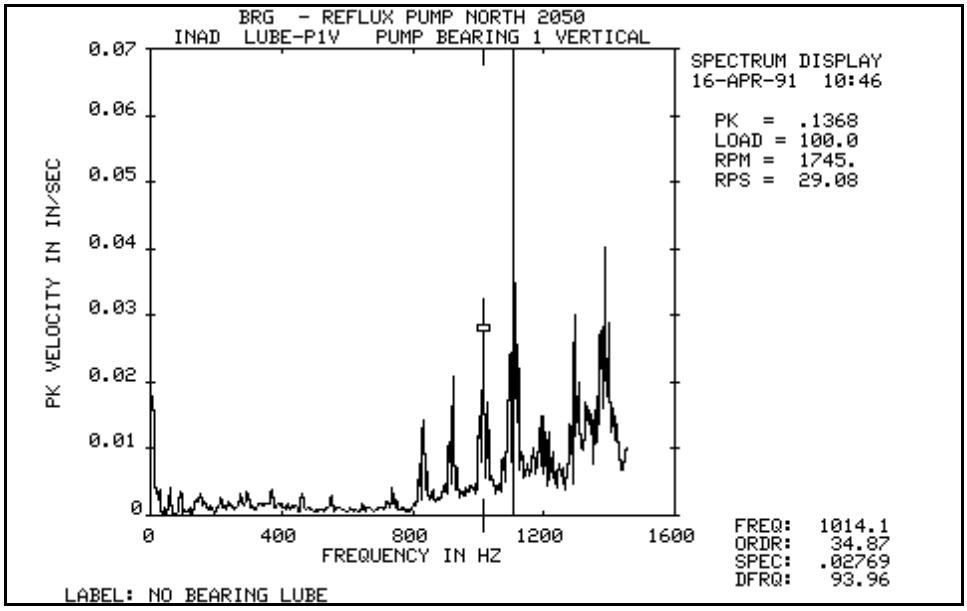
Bearing Case #7



The separation of the peaks is spaced at around 93 Hz. Typically the spacing between the peaks are around 80 - 130 Hz.

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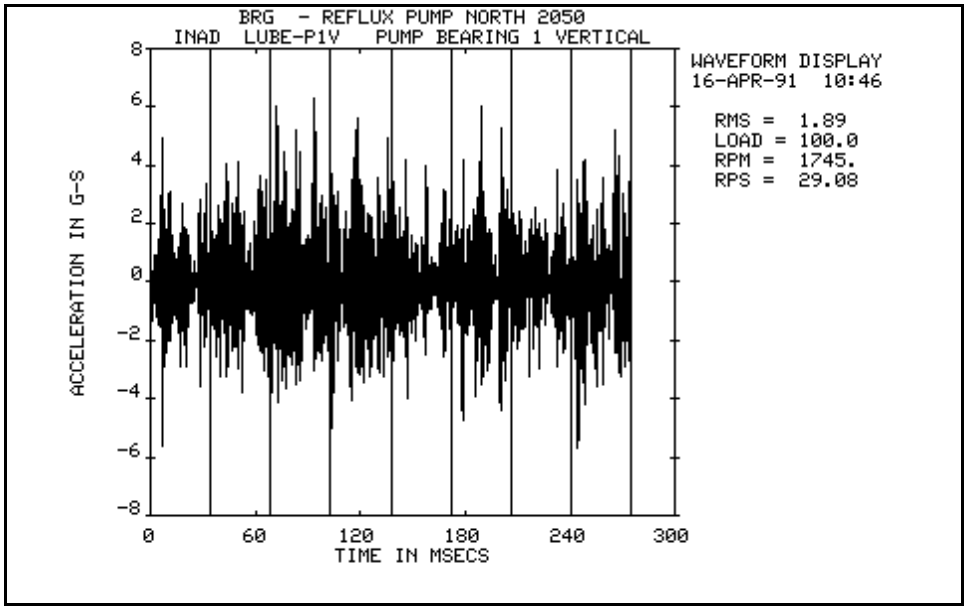
Bearing Case #7



The waveform displayed above shows significant impacting in the waveform.

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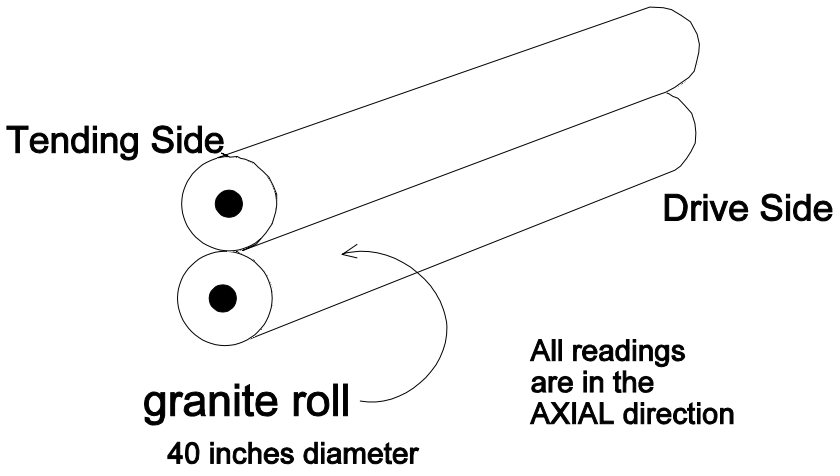
Bearing Case #8



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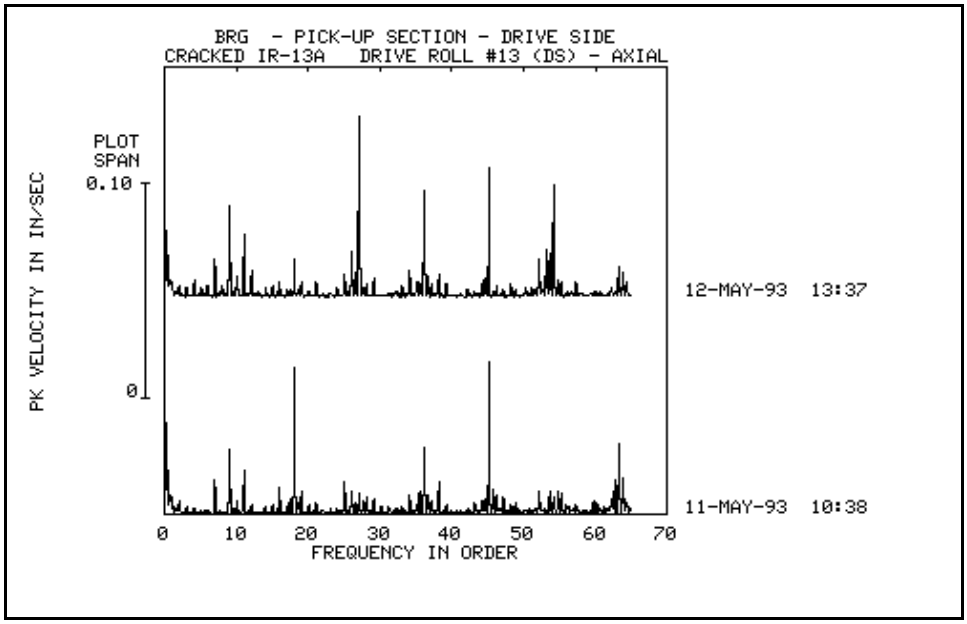
Bearing Case #8



The multiple spectra plot shown above shows data taken on the same point, one day apart. It appears that sidebands are present around the higher frequency peaks. The following pages examine the single spectrum closer.

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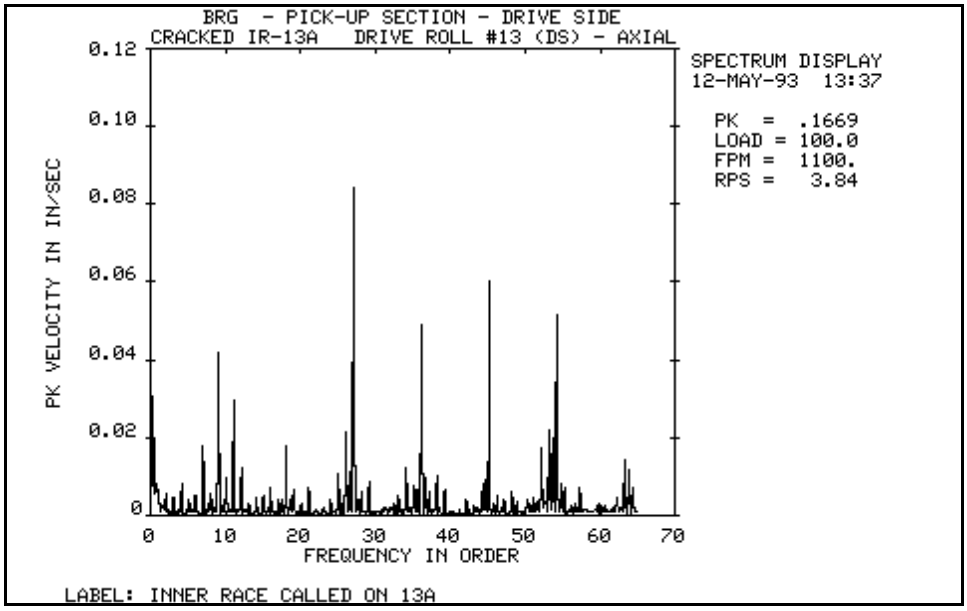
Bearing Case #8



The spectrum shown above is a full screen of the May 12, 1993, data. The peak list on the following page shows that most of the energy in the spectrum is nonsynchronous energy.

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Bearing Case #8 Peak List

```
Machine:       (BRG ) PICK-UP SECTION - DRIVE SIDE
Meas. Point:   CRACKED IR-13A  -->  DRIVE ROLL #13 (DS) - AXIAL
Date/Time:     12-MAY-93   13:37:34     Amplitude Units:  IN/SEC  PK
Data Label:    Inner Race Called on 13A
```

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE	PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	.94	.0308	.24	13	104.01	.0911	26.42
2	1.66	.0194	.42	14	130.84	.0129	33.23
3	2.04	.0147	.52	15	138.68	.0496	35.22
4	26.83	.0181	6.81	16	141.00	.0102	35.81
5	34.67	.0433	8.81	17	146.53	.0104	37.22
6	38.59	.0099	9.80	18	173.35	.0605	44.03
7	42.51	.0323	10.80	19	200.18	.0175	50.84
8	46.42	.0122	11.79	20	204.10	.0230	51.84
9	69.34	.0203	17.61	21	205.71	.0202	52.25
10	96.17	.0124	24.43	22	208.03	.0551	52.84
11	100.09	.0216	25.42	23	242.70	.0167	61.64
12	101.71	.0116	25.83	24	245.01	.0118	62.23

```
TOTAL MAG     SUBSYNCHRONOUS       SYNCHRONOUS      NONSYNCHRONOUS
   .1669        .0371  /  5%       .0748  / 20%       .1445  / 75%
F1/Enter=Accept      F2=Paging is OFF      F7=Title       F9=Copy       Esc=Quit
```


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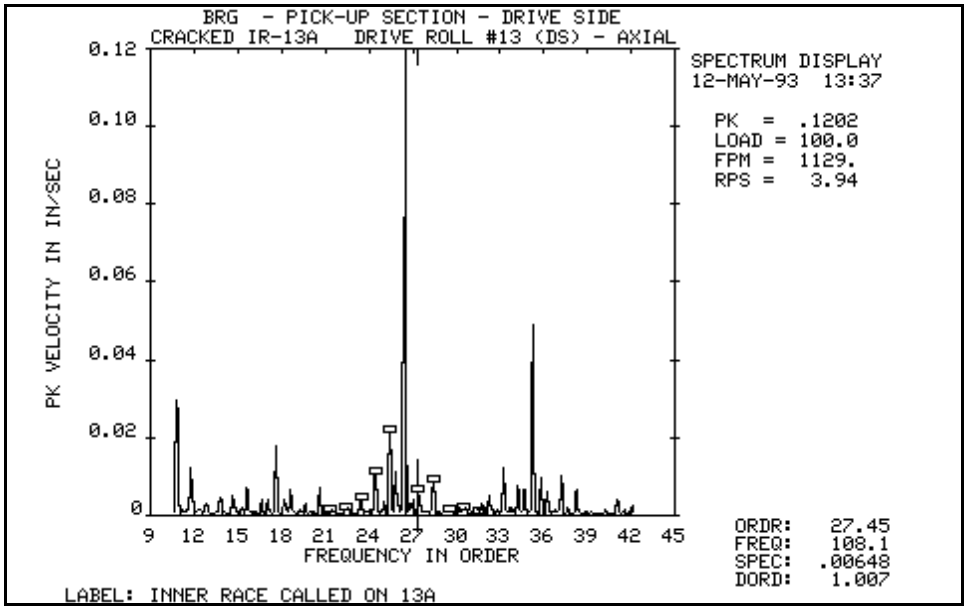
Bearing Case #8



As shown in the spectrum above, sidebands spaced at 1xTS are present. The characteristic of the sidebands spaced at 1xTS around a primary frequency is associated with an inner race defect. Amplitudes are low but significant numbers of harmonics and sidebands are present in the spectrum.

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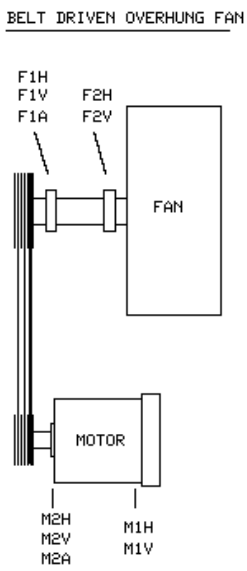
Bearing Case #9
530 Baghouse Fan
Belt Driven, Overhung



Notes:

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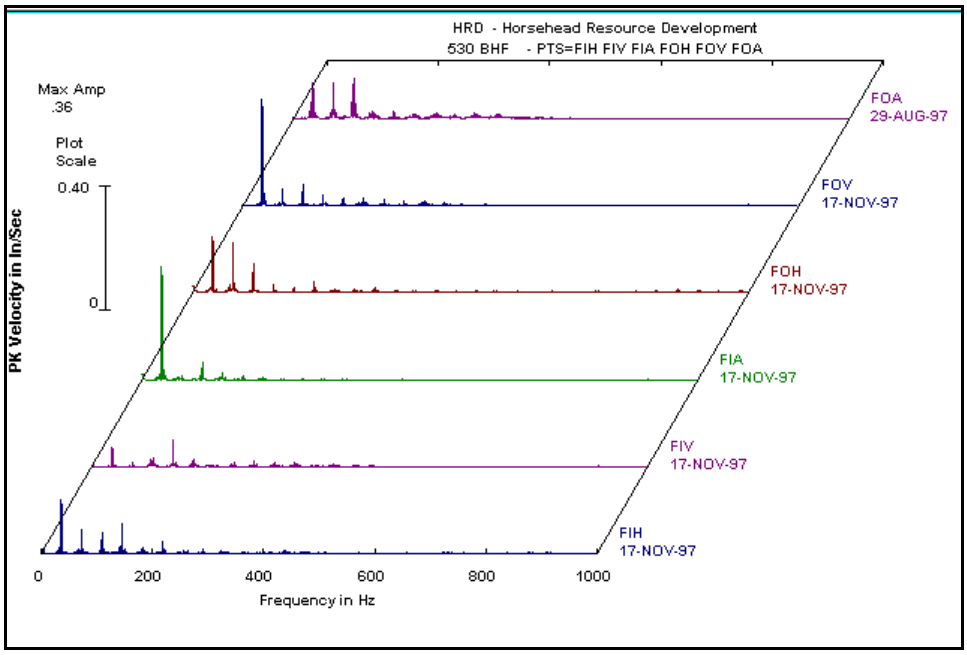
Bearing Case #9

Displayed is a multiple points plot. There appears to be a problem on both ends of the fan, inboard and outboard.



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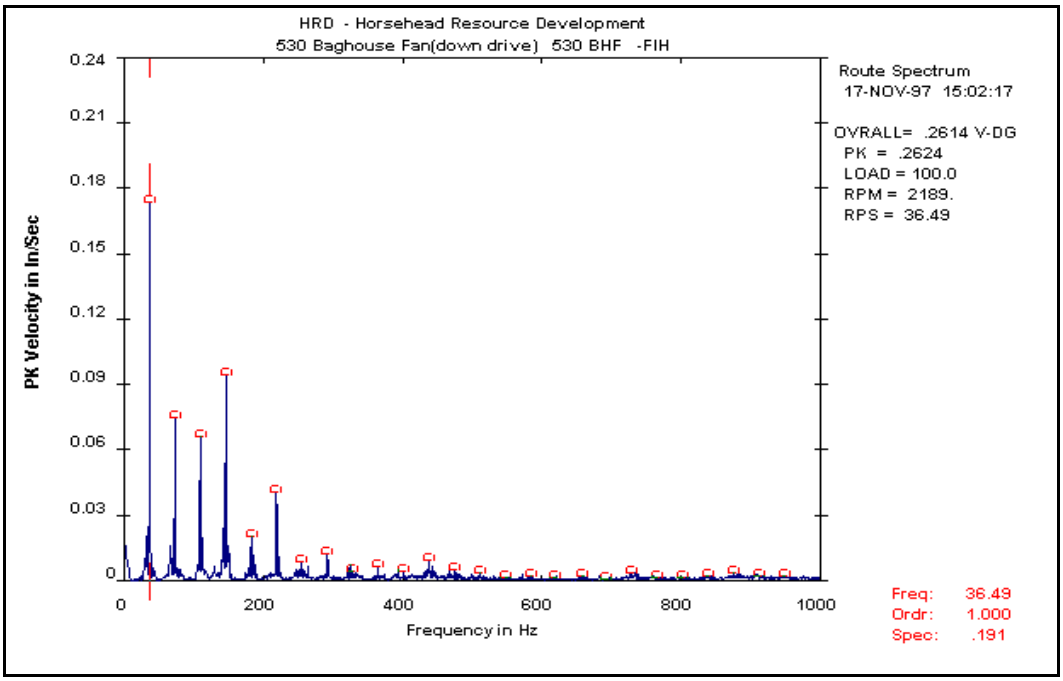
Bearing Case #9

Full scale plot of FIH reveals several harmonics of runspeed. This could be an indication of a Looseness condition on the machine. Structural or Component Looseness ?



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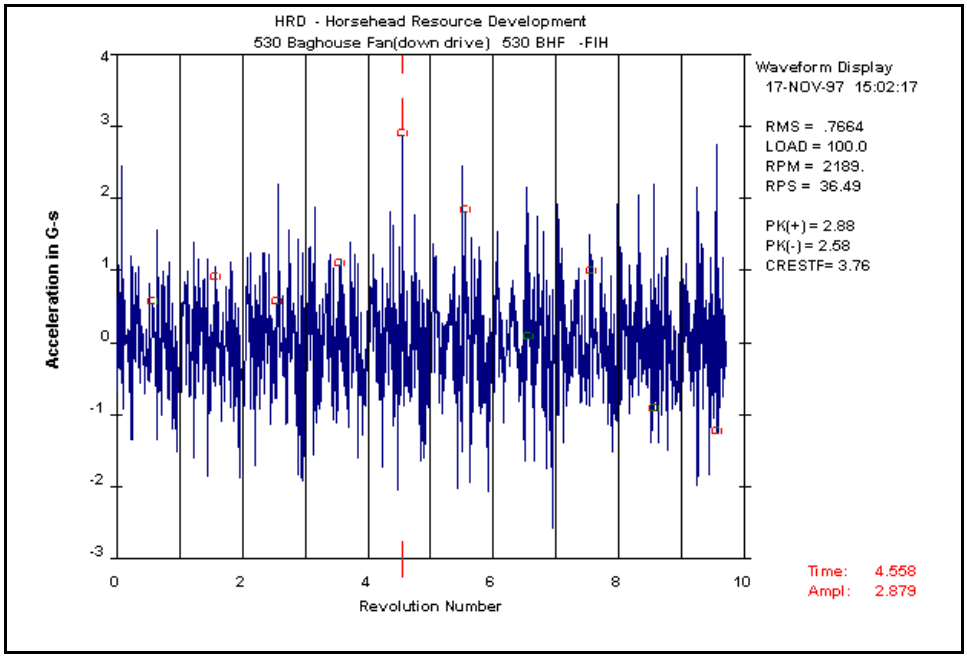
Bearing Case #9

There is a lot of impacting in the Waveform on FIH, 5 to 6 G swing. The pattern of the impacting looks non-repetitive. What would this lead you to assume as to the type of problem we have on this one? The Crest Factor is 3.76.



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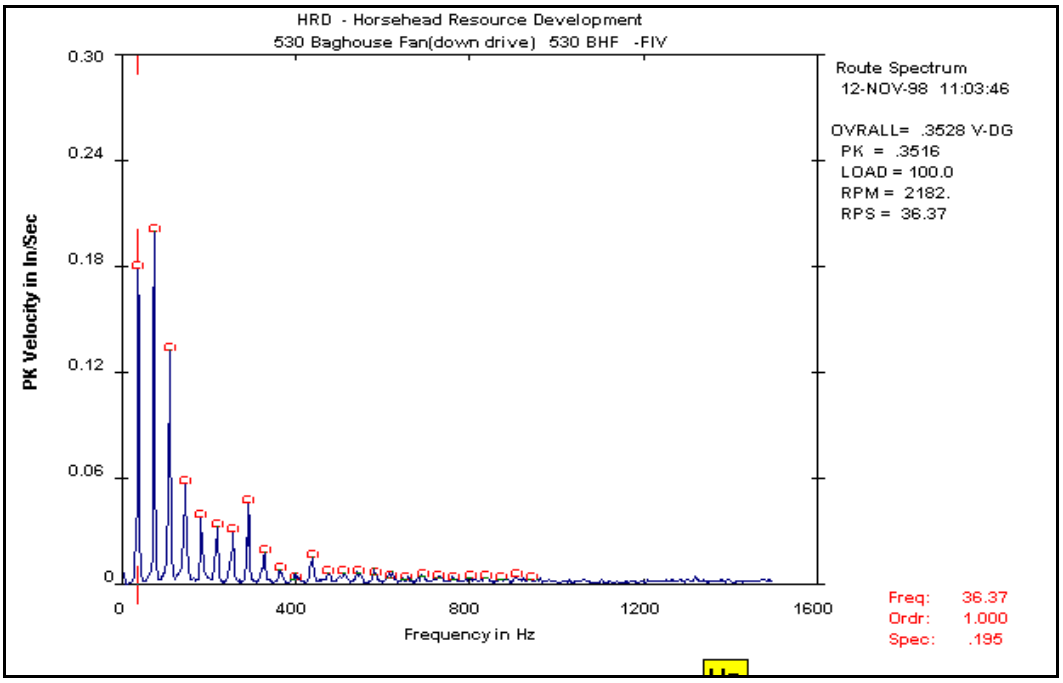
Bearing Case #9

Before you make a decision on this one, let's take a look at the data collected in the Vertical direction FIV. Again there are a lot of harmonics of runspeed in the spectrum. Looseness?



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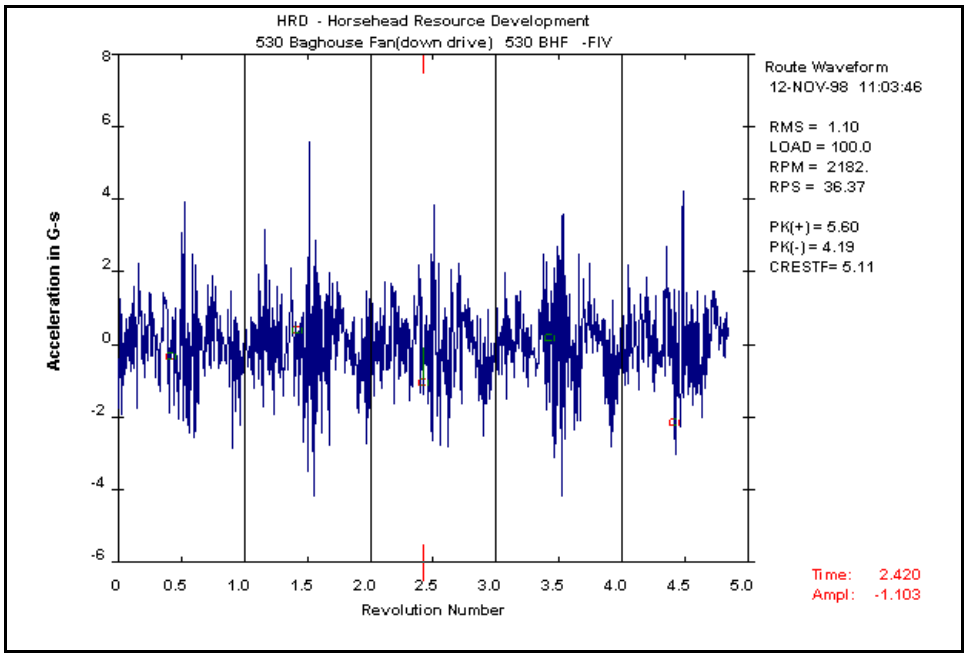
Bearing Case #9

Let's take a look at the Waveform for FIV. Notice in the vertical direction the impacting is more of a repetitive type pattern. There is a 10 G swing and also the crest factor is 5.11. Now what type problem do we have on this fan? If you said bearing looseness, you are correct.



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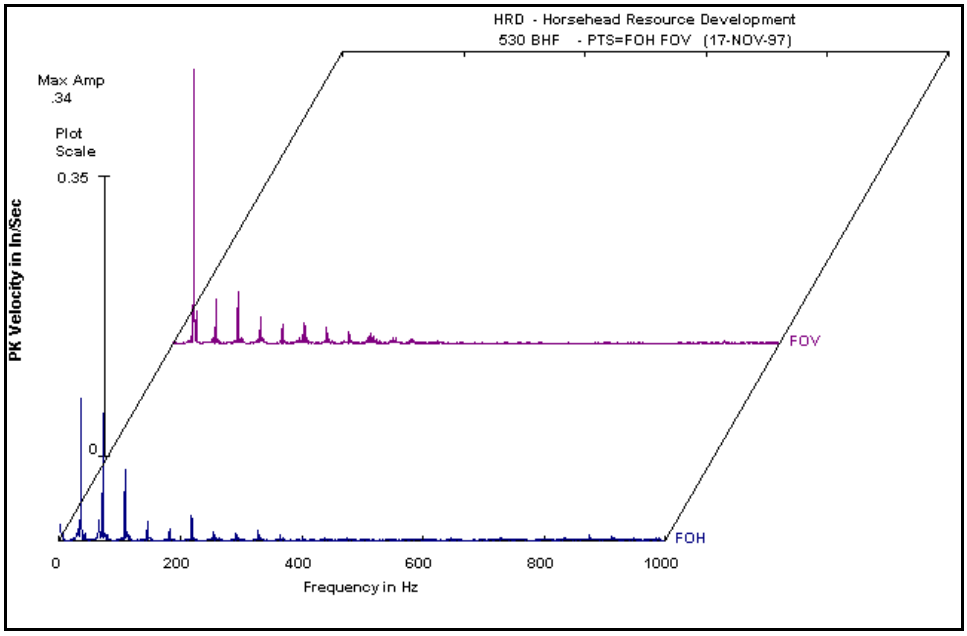
Bearing Case #9

Displayed in this plot is the horizontal and vertical directions of the outboard end. There are again a lot of harmonics of runspeed.



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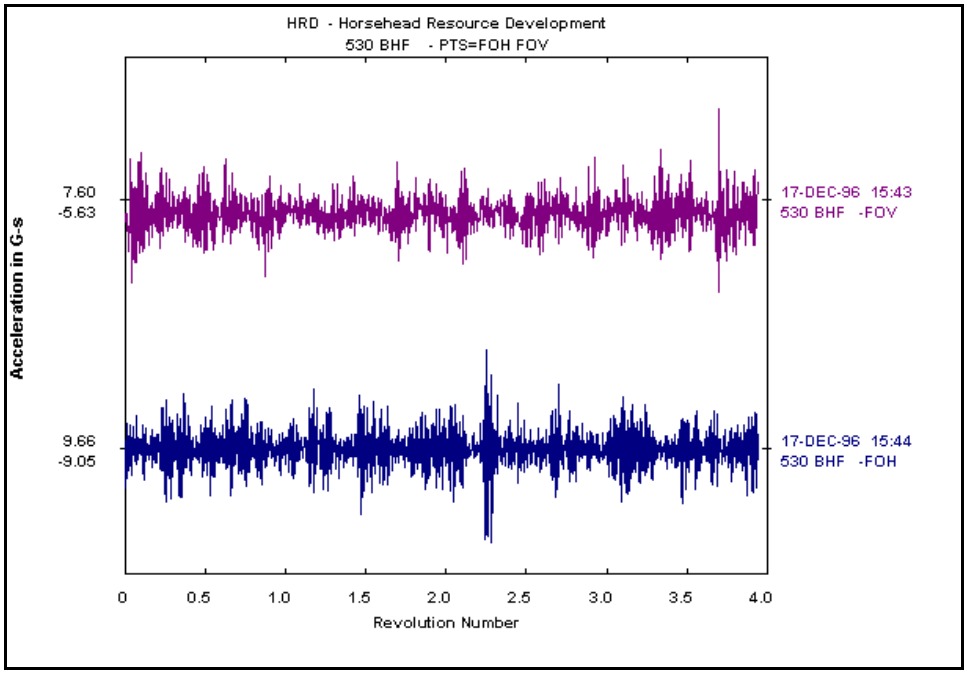
Bearing Case #9

When we compare the waveform patterns of the outboard horizontal and vertical readings we should be able to determine what type of problem we have on this bearing.



The horizontal direction on this bearing indicates we have a bearing looseness problem. When the bearings were changed they found both to be very worn with defects on balls, inner and outer races and very loose.

Note: It pays to look at data taken in all directions when trying to determine a problem on a machine.

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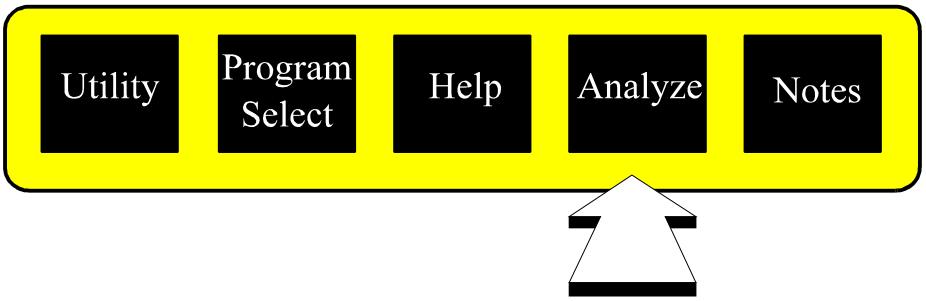
Section Eight
Data Averaging

Objectives

- Use MasterTrend to set up different averaging methods.
- Describe each of the averaging modes available in the Model 2120.
- Determine specific applications for each of the averaging modes.

General Description

Each of the averaging modes lets you perform in-depth analysis in the field as the need arises. The following lessons discuss each of the averaging modes in detail. First we will show how to setup MasterTrend to collect route data with normal as well as advanced averaging techniques. We will also show the use of the analyze function to collect field data.



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Averaging Modes

(Valuable tools for Analytical Trouble Shooting)

Normal Averaging — Normally used for routine data collection and when other averaging modes are inappropriate. Normal averaging enhances repetitive events. ***--- no tach is needed ---***

Peak-Hold Averaging — Use to measure a machine that varies in speed (frequency) when you do not have a reference pulse (e.g., during coast down). You may also use this mode to monitor random fluctuations such as steam whirl in a turbine or oil whirl in a journal bearing. ***--- no tach is needed ---***

Synchronous Time Averaging — Use to minimize all vibration not synchronously related to the shaft providing the reference pulse. All vibration at a whole number order of shaft speed (frequency) remains in the spectral data. Amplitudes at nonsynchronous frequencies are reduced, virtually eliminating vibration from other components on the same machine.
--- tach is needed ---

Order Tracking — Use to monitor machinery that continuously varies in speed during the data collection process. This method requires a reference pulse. ***--- tach is needed ---***

Negative (Linear Minus) — Dynamically subtracts two spectra. This method is one of the most powerful analysis tools available, because it subtracts data while the machinery is in operation. ***No tach is needed.***

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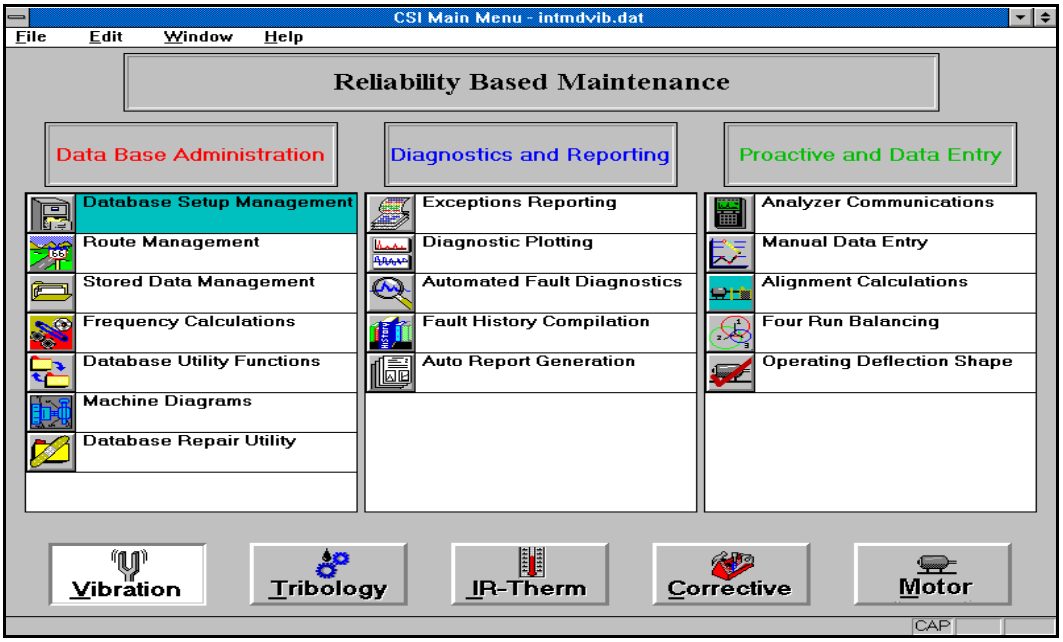
MasterTrend Setup - Data Averaging

All the data averaging methods listed on the previous page, with the exception of **Negative Averaging**, are accessible in the Analysis Parameter (AP) Sets in MasterTrend. Let's go to **INTMDVIB.DAT** to see the setup.

Select **Database Setup Management**.



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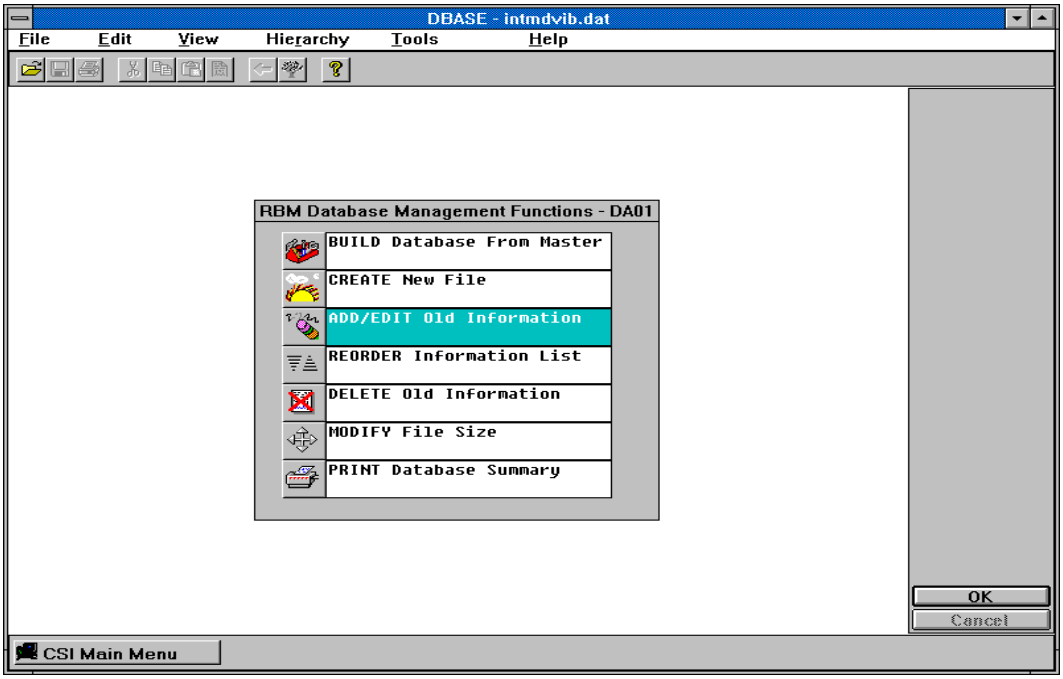
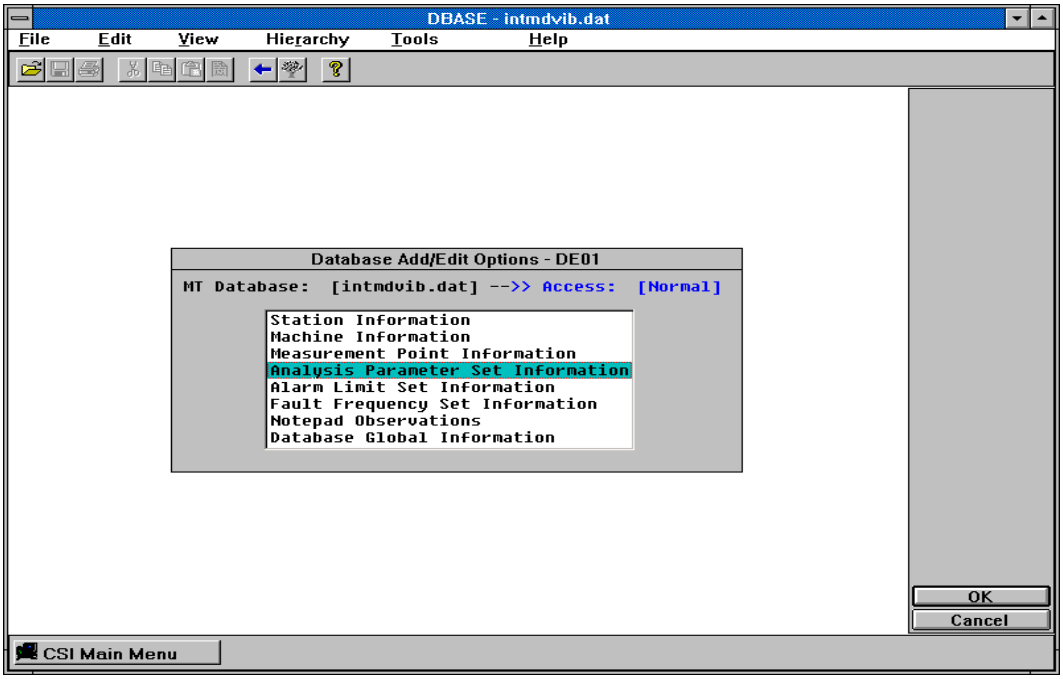
Then select **ADD/EDIT Old Information**.



We will look at the AP sets for the various averaging modes.



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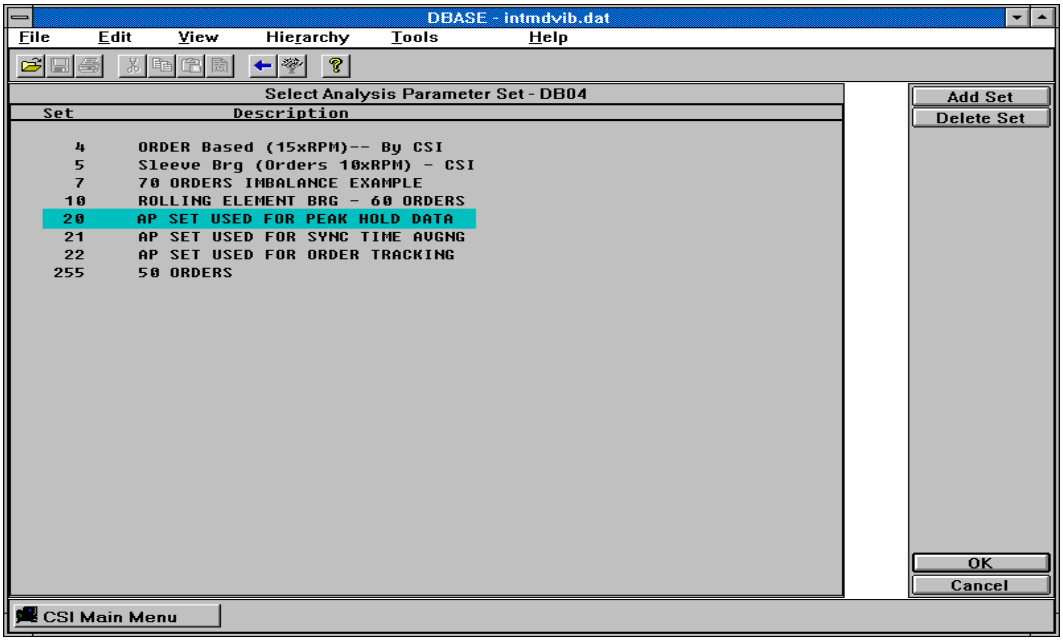
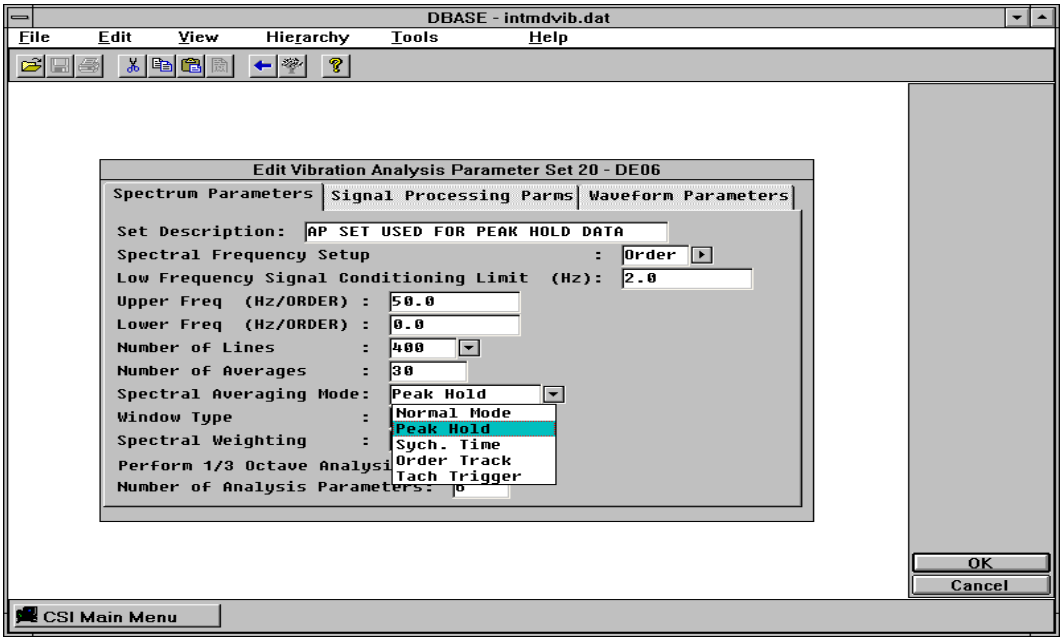
The first averaging mode is **Peak-Hold**.



The choice of averaging methods is made in the **Spectral Averaging Mode**. The rest of the choices are shown.



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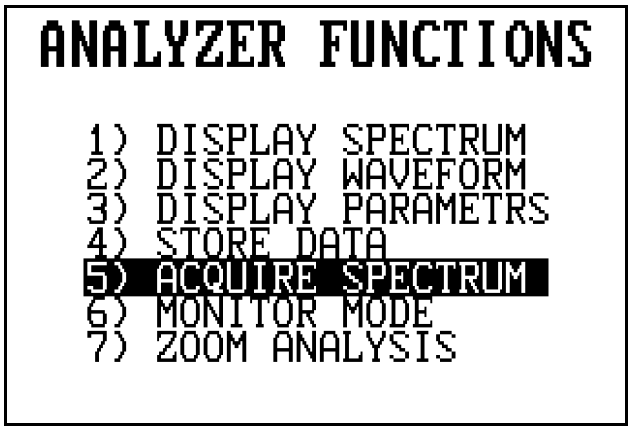
Acquire Spectrum - Lab

We can also access the various averaging methods from the **Analyze, Acquire Spectrum** function. This lab will explain each Analyze function using the analyzer.

Normal Averaging (also called linear averaging) uses a specific number of averages — usually four to eight — that carry equal weight. The two types of normal averaging are linear plus and linear minus. The linear plus averaging mode is more common. Your routine data collection normally uses this averaging mode. Linear plus proves most useful for attenuation of nonrepetitive events.

Press the **Analyze** function key on the analyzer to display the Analyze Functions menu as shown below. Highlight **5) ACQUIRE SPECTRUM** and press **Enter** to define the menu for data collection.



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Define page one of the three-page display as shown below. When you complete page one, press the **Page Up** key to advance to the second page of the measurement definitions.

```
ACQUIRE SPECTRUM

FREQUENCY  : 400        HZ
LOW CUTOFF: 2.00        HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: VELOC/DIGITL
INIT SETUP:  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```

Normal averaging is the most commonly used for data collection. In this mode, the analyzer stores the amplitude measured for each ***cell*** (bandwidth). The process then adds the next sample and divides the results by two. The analyzer gives each succeeding sample the same weight. Define your menu as shown below and press **Enter** to proceed with data collection.

```
ACQUIRE SPECTRUM

AVERG MODE: NORMAL
TRIG MODE :  OFF
TRIG LEVEL: 0.00
% PRE TRIG: 0
FS RANGE  : 0.00
A-WEIGHT  :  NO
TRIAX CTRL: OFF

PAGE 2 OF 3   USE PAGE KEYS TO ADVANCE
```


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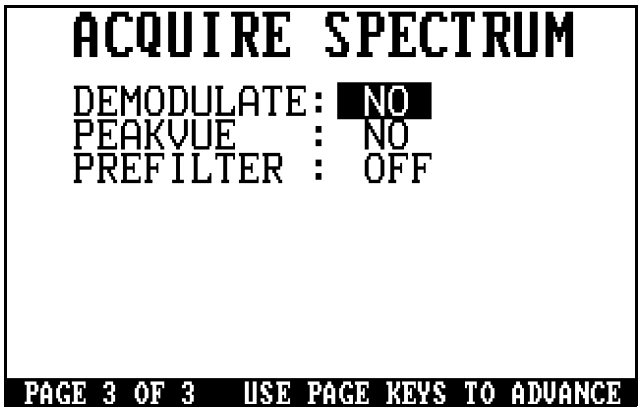
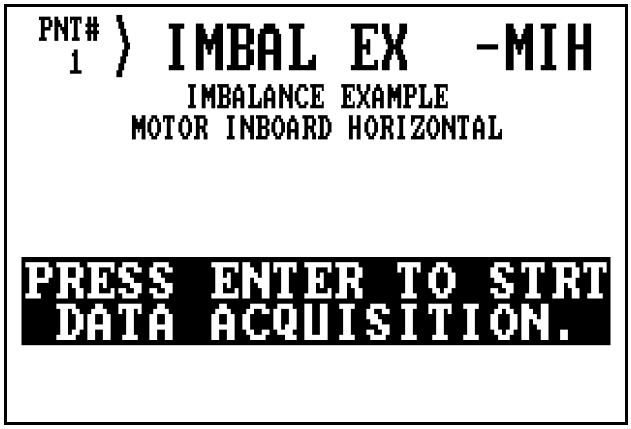
The third page of the Acquire Spectrum Menu is shown below. It contains the option to choose demodulation or PeakVue analysis and the filter settings for either. Press **Enter** to accept page 3.



The following screen displays. When ready to collect data, press **Enter** once again to begin data collection.



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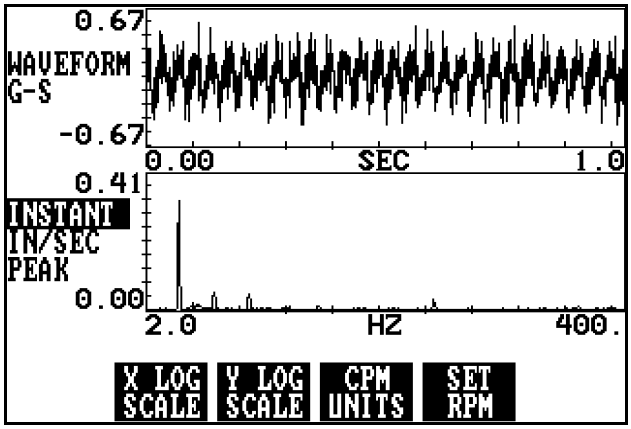
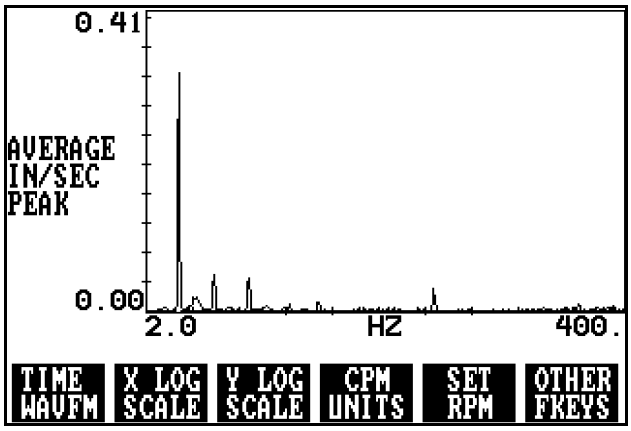
What the data being collected might look like is shown below. To display these plots, press the **Utility** key on the analyzer and select **DUAL DISPLAY**.



A plot displaying the full view of the spectrum displays after the analyzer completes the data collection.



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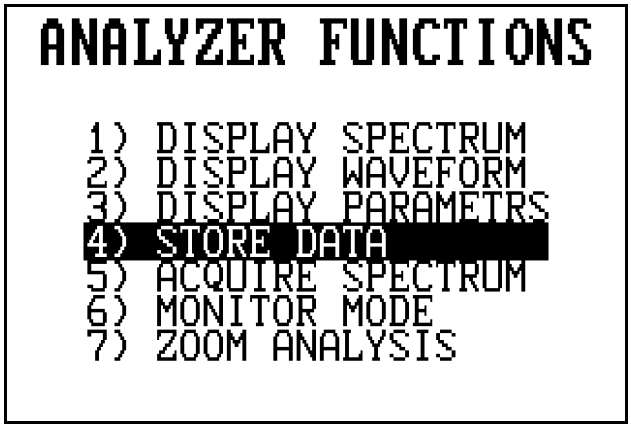
Press **Enter** to return to the Analyzer Functions Menu. Highlight **4) STORE DATA** and press **Enter**. The data will be stored at the measurement point that was active when the Analyze function key was pressed.



Select the option that best suits the data that you actually want to save. Select **3) STORE BOTH** and press **Enter**.

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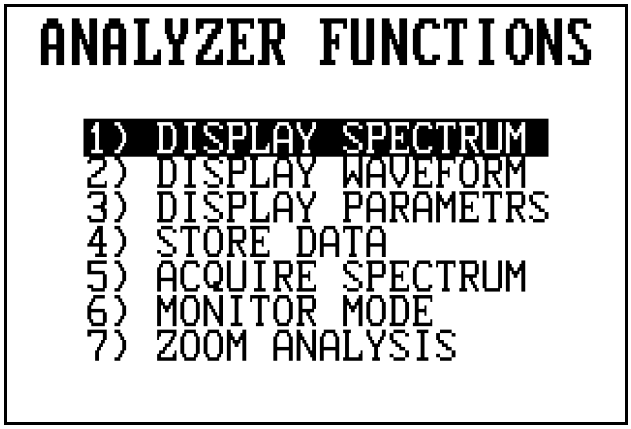
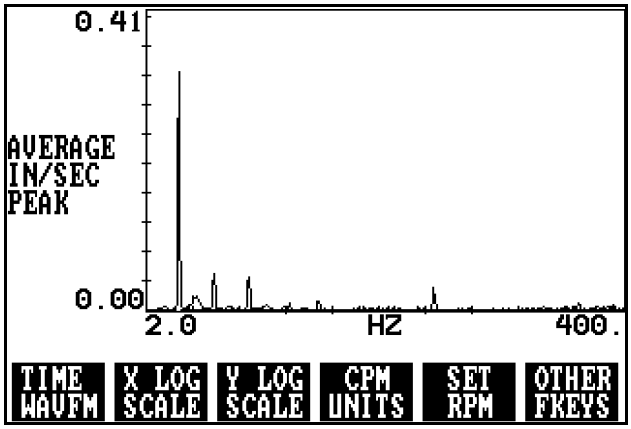
To view saved data, highlight **1) DISPLAY SPECTRUM** and press the **Enter** key.



The spectrum will be displayed. Note the different function keys available. The waveform can be accessed directly and the frequency units may be changed easily.



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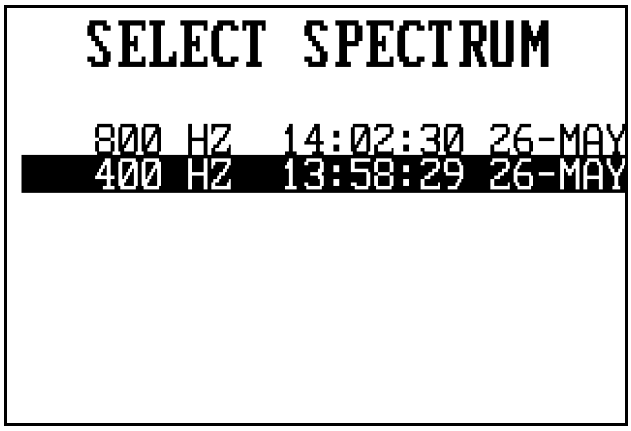
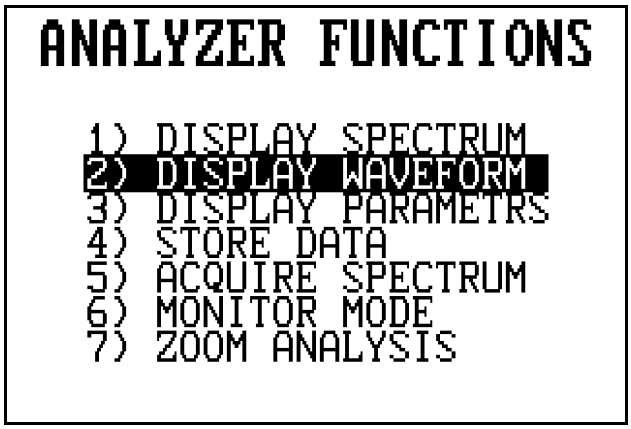
If more than one spectrum has been stored, you must select the spectrum to view.



The waveform may be displayed directly by selecting this option from the Analyzer Functions Menu.



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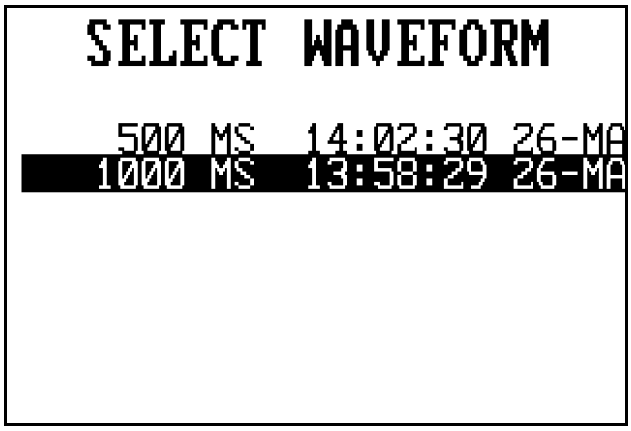
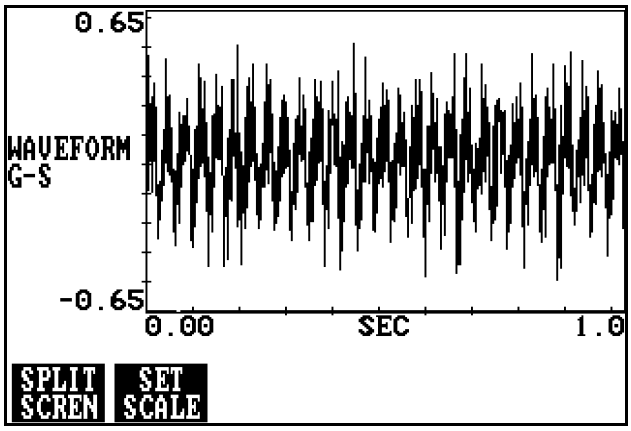
In this case, multiple waveforms have been saved. Notice that the length of the waveform displays. Highlight the waveform you want to display and press **Enter**.



The waveform displays. The function keys allow for various operations.



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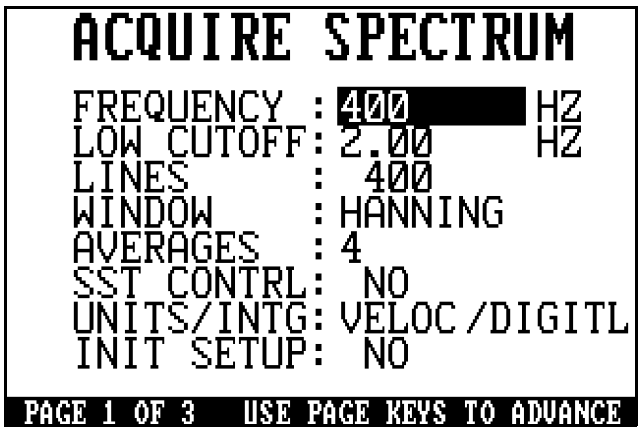
Acquire Spectrum - Definition

Let us examine each of the acquire spectrum options more closely. Press the **Analyze** function key at the top of the analyzer. Select the **ACQUIRE SPECTRUM** option. **PAGE 1 of 3** is the first screen to discuss. The Frequency selection should be set above the highest frequency of interest. This is the F_{max}.



Frequencies below the Low Cutoff will not be included in the calculation of the overall measurement nor will they be displayed in the spectrum. This value is often set at a frequency equal to two times the F_{max} divided by the number of lines.

$$\textit{Low Cutoff} = \frac{2 * \textit{Frequency}}{\textit{Number of Lines}}$$

Note: 2 Hz is a good selection for Low-End Cutoff for most data collection

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```
ACQUIRE SPECTRUM

FREQUENCY  : 400      HZ
LOW CUTOFF: 2.00      HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: ACCEL/ANALOG
INIT SETUP:  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```

The lines of resolution divided into the frequency result in the ***Bandwidth*** (BW), sometimes referred to as ***Resolution***. Choices in the 2120 analyzer are 100, 200, 400, 800, 1600, 3200, 6400, and 1/3 Octave.

$$\textit{Bandwidth} = \frac{\textit{Frequency}}{\textit{Number of Lines}}$$

Two window choices are available — Hanning and Uniform. The Hanning window is selected most often because it is best suited to accurate frequency analysis. The Uniform window is usually reserved for transient analysis.

```
ACQUIRE SPECTRUM

FREQUENCY  : 800      HZ
LOW CUTOFF: 4.00      HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: ACCEL/ANALOG
INIT SETUP:  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```


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The number of averages selected is typically set between 4 and 8 for normal data collection. The actual values selected depend most on the averaging mode.

```
ACQUIRE SPECTRUM

FREQUENCY  : 800      HZ
LOW CUTOFF: 4.00      HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: ACCEL/ANALOG
INIT SETUP:  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```

The SST Control set to YES allows for low frequency data collection with an amplitude correction applied to the low frequency data. Leave set to NO for now.

```
ACQUIRE SPECTRUM

FREQUENCY  : 400      HZ
LOW CUTOFF: 2.00      HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: VELOC/DIGITL
INIT SETUP:  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```

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The **Units / Integration** option allows the analyst to select the spectral units and the integration mode without having to go back to the Utility function key / Change Setup.

```
ACQUIRE SPECTRUM

FREQUENCY  : 800      HZ
LOW CUTOFF: 4.00      HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: VELOC /DIGITL
INIT SETUP:  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```

```
ACQUIRE SPECTRUM

FREQUENCY  : 800      HZ
LOW CUTOFF: 4.00      HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: VELOC /DIGITL
INIT SETUP:  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```


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The Initial Setup selection set to YES resets the Acquire Spectrum choices to the values used in the current Route Point. Leave this set to NO.

```
ACQUIRE SPECTRUM

FREQUENCY  : 400      HZ
LOW CUTOFF: 2.00      HZ
LINES      :  400
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL:  NO
UNITS/INTG: VELOC/DIGITL
INIT SETUP:  NO
PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```

Press the **Page Up** key to move to the second page of the Acquire Spectrum Menu. In this case, the averaging mode selected is Peak Hold. The options on this page will be discussed later as they become relevant to the measurement being made.

```
ACQUIRE SPECTRUM

AVERG MODE: PK HOLD
TRIG MODE :  OFF
TRIG LEVEL: 0.00
% PRE TRIG: 0
FS RANGE  : 0.00
A-WEIGHT  :  NO
TRIAX CTRL: OFF

PAGE 2 OF 3   USE PAGE KEYS TO ADVANCE
```

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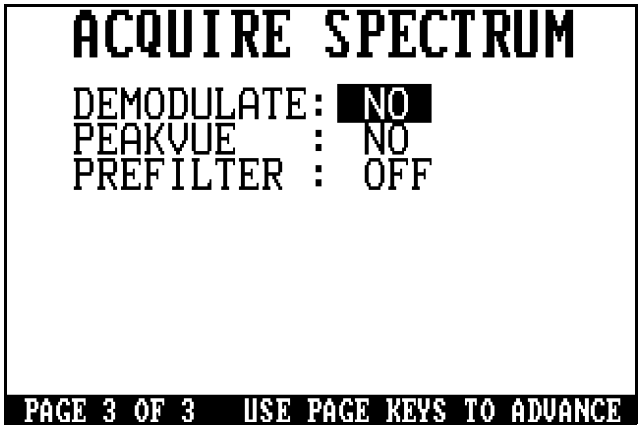
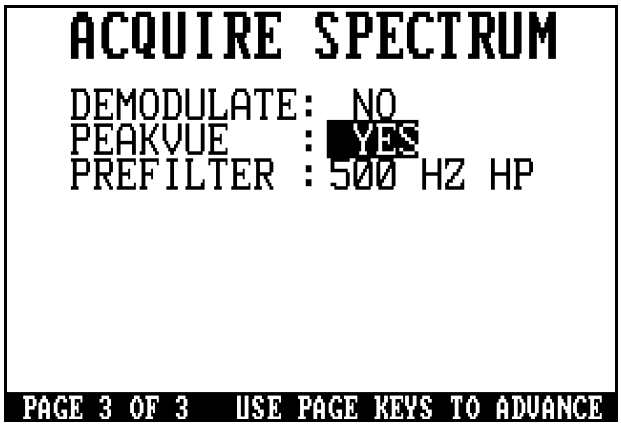
Press the **Page Up** key to move to page 3 of 3. When Demodulate is set to YES, the analyzer's built-in demodulator is activated for the measurement. When enabled, the filter option is activated, allowing a choice of 500, 1000, 2000, 5000, and 10,000 Hz high pass filters or a 500 to 1000 Hz band pass filter. These filters remove the signal below the setpoint, amplify the remaining signal, and then demodulate it, resulting in low-frequency data up to 40 % of the filter set point in Hz.



Toggling the PeakVue option to YES, turns on PeakVue analysis. This allows spectral analysis of very short duration spikes in the data typically missed in normal spectra. When the PeakVue option is on, the prefilter option will automatically enable, allowing different filters to be chosen.



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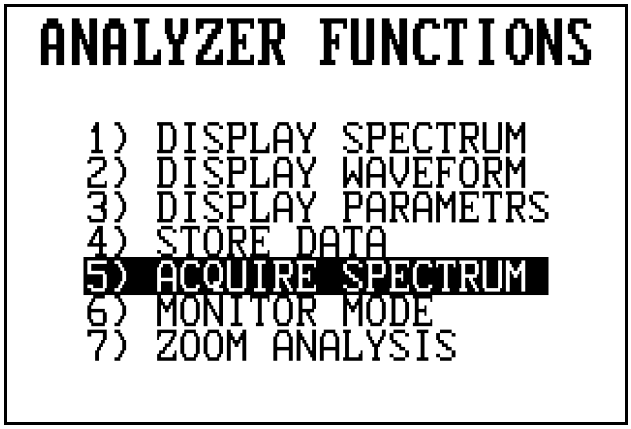
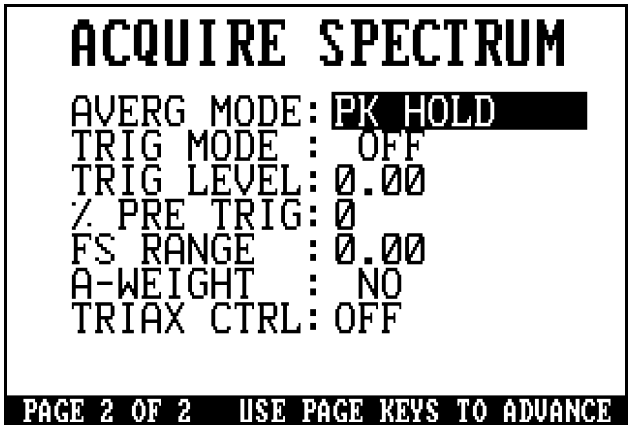
Peak-Hold Averaging

Peak-hold averaging retains the highest amplitude at every frequency from all the averages acquired. It is most commonly used for coastdown data for all frequencies when there is not a once-per-revolution (tach) event marker available. Peak-hold averaging can also be used when amplitudes are unstable from sample to sample. Press the **Analyze** function key to display the Analyze Functions Menu. Highlight **5) ACQUIRE SPECTRUM** and press **Enter**.



Peak-hold averaging retains the highest value for every spectral line. It uses these values to develop an envelope that displays the highest amplitude measured at each frequency during the entire sampling process.



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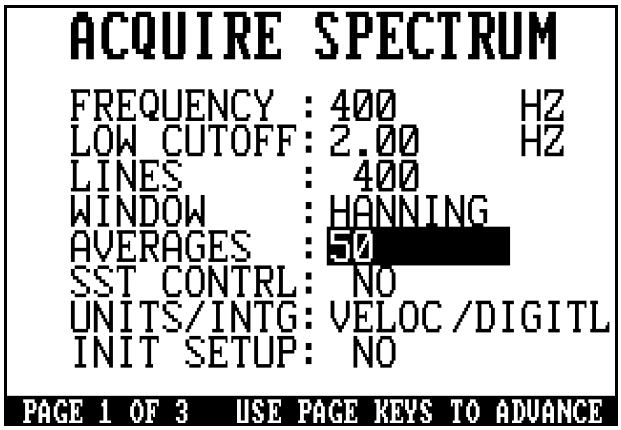
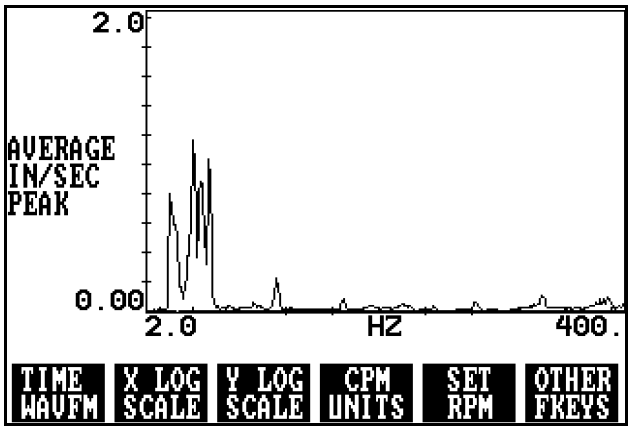
The number of averages should be set high. Use peak-hold averaging when the vibration signal varies over a period of time. Peak-hold can also be used when random fluctuations take place during the data collection period. Press **Enter** to initiate data collection. Press **Enter** again when you are ready to collect data.



The spectrum shown below illustrates peak-hold averaging. The highest amplitude that occurred during the averaging process is retained at each frequency.



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Synchronous Time Averaging

Synchronous time averaging collects time waveforms synchronized to a tachometer (tach) pulse. The synchronous time waveforms average in the time domain. The resulting spectral display is the FFT for the averaged waveform. Use synchronous time averaging when you need data directly related to the turning speed of a specific shaft. For example, use synchronous averaging when you have several machines (or shafts on one machine) that all turn at different speeds. It is important to remember that many faults generate nonsynchronous energy and they will be reduced with this averaging. This averaging will enhance turning speed and harmonics of turning speed.

Note: Synchronous averaging reduces but does not eliminate nonsynchronous vibration in the spectral data. The amount of reduction depends on the number of averages you take. Use the formula below to calculate the results:

$$\textit{Reduction Factor} = \sqrt{\textit{Number of Averages}}$$

For example, 100 averages reduce the nonsynchronous vibration by a factor of 10, and 10,000 averages by a factor of 100.

It is important to understand that the energy averaged out using ***Synchronous Time Averaging*** can be sub-synchronous, synchronous, or nonsynchronous energy from another source or machine that you are ***not*** collecting data on. Also, sub-synchronous and nonsynchronous energy will be reduced or eliminated from the spectrum that you ***are*** collecting the data on. Only the synchronous energy that is occurring on the machine you are collecting data on using ***Synchronous Time Averaging*** with a photo-tach will remain in the Spectrum.

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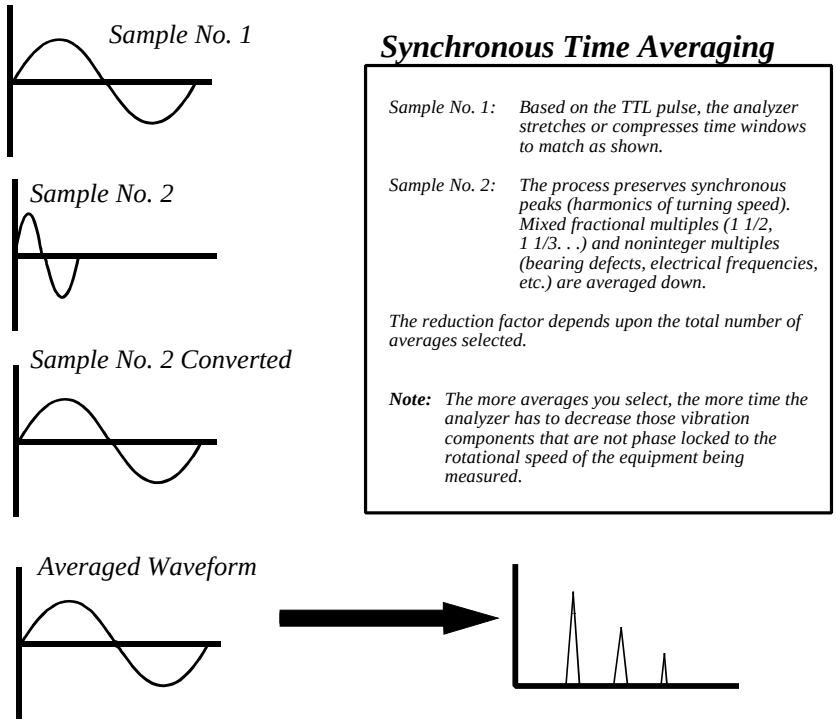
Synchronous vs. Harmonic

Vibration signals at integer multiples of a primary frequency (any frequency where a peak in the signal may occur) are called harmonics of that frequency.

- Harmonics = N x f, where f is the primary frequency, and N = 1, 2, 3, etc.
- Synchronous peaks occur at integer multiples of the operating frequency of the shaft at the measurement point (1xTS, 2xTS, 3xTS. . .).



Synchronous peaks are harmonics of shaft frequency. You may also have harmonics of nonsynchronous peaks as in the case of roller bearing defect frequencies, but these peaks are **not synchronous**.

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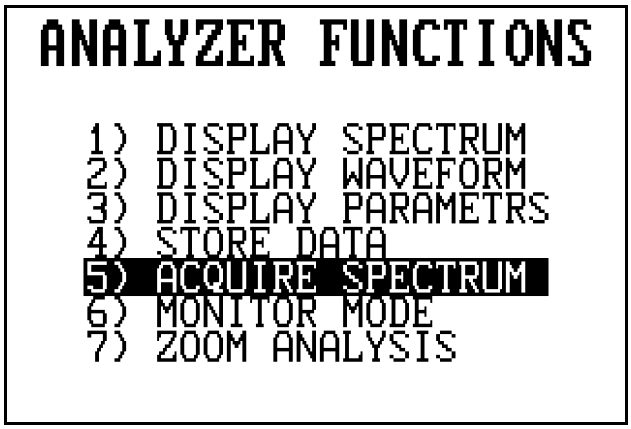
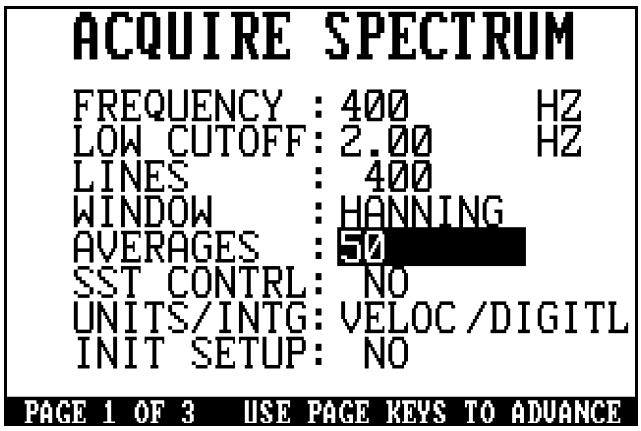
Press the **Analyze** function key to display the Analyzer Functions Menu. Highlight **5) ACQUIRE SPECTRUM** and press **Enter**.



Define the three-page menu as shown on the screens here. On page one, specify **50** averages. The analyzer reduces the nonsynchronous energy by the square root of 50, a reduction factor of 7.07.



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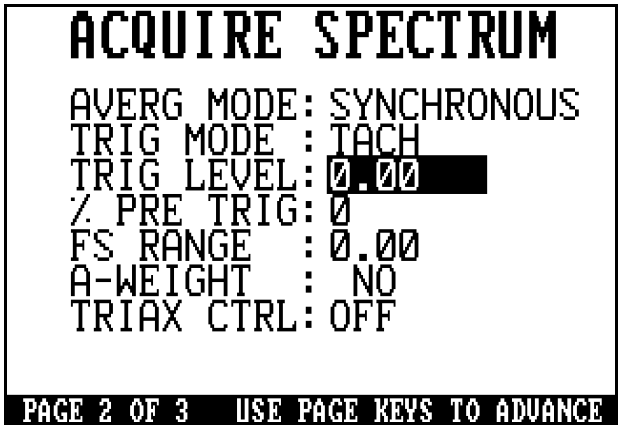
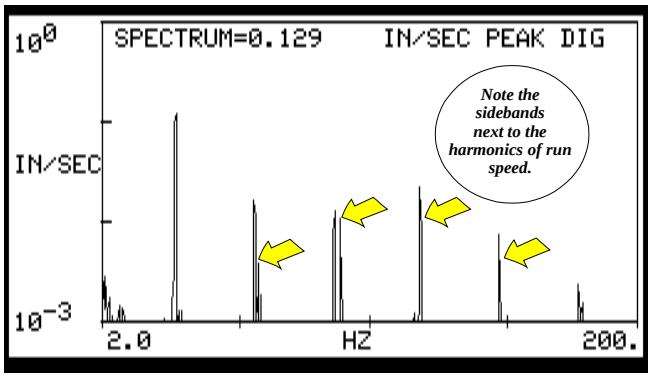
Press the **Page** key to advance to page two of the menu. Toggle Average Mode to **SYNCHRONOUS**. The analyzer trigger mode must be toggled to **TACH**. A tach signal is required to average this data.



Press **Enter** to begin the data collection. The analyzer automatically displays the averaged spectrum when collecting data. After completing the collection period, the analyzer displays the final average. The spectral display below was collected with normal averaging. Note the sideband energy. Even though those sidebands indicate a defect frequency, the synchronous time average data eliminates them from the plot.



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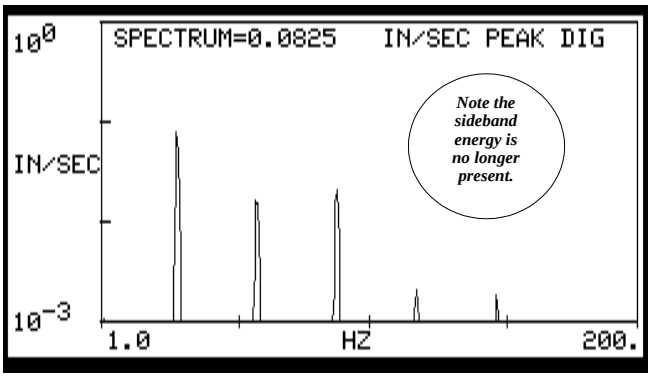
The spectrum shown below is the same machine seconds later. With the synchronous time averaged data, the sidebands and all other nonsynchronous data almost disappear.



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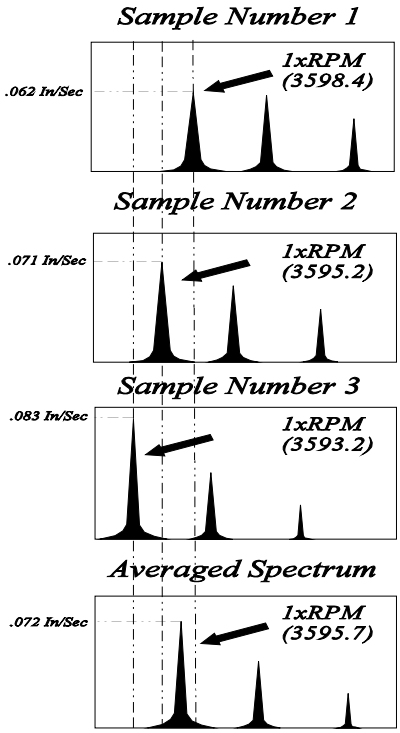
Order Tracking

Order tracking automatically adjusts the analyzer during data collection to account for the changes in machine turning speed (RPM) using a reference tach pulse. Use this function primarily on variable speed machines.

When a machine's speed changes, the peaks associated with certain faults often get smeared over many frequencies on the spectral display. It may be difficult to determine what peak corresponds with 1xTS, 2xTS, etc., making it impossible to identify bearing fault frequencies, belt frequencies, and other characteristic fault frequencies. Order tracking, however, automatically adjusts for changes in the machine RPM to prevent smearing these peaks.

The illustration below shows how order tracking averages spectral data. With a total of three averages and a speed variance of less than 6 percent between samples, the order tracking keeps up with both amplitude and speed. Once the data collection finishes, the analyzer averages the data together in both the X-axis (frequency) and Y-axis (amplitude). Order tracking also stretches or compresses the size of the time window so that all windows match.



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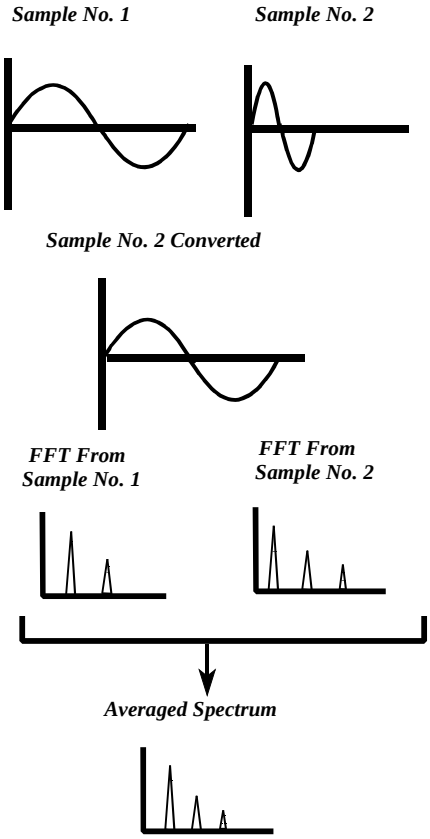
Order Tracking Averaging

When machine speed varies, peaks related to turning speed tend to smear and become reduced during normal averaging. Order tracking retains any persistent signal related to turning speed, including frequencies where bearing faults occur.

- *The analyzer stretches or compresses the size of each time window so that all sample windows match.*
- *The analyzer performs an FFT on each sample.*
- *The resulting spectra are averaged in both the Y axis (amplitude) and the X axis (frequency).*
- *The end result is an averaged spectrum with both synchronous and nonsynchronous data, with all the data displayed relative to the averaged turning speed (1st Order).*



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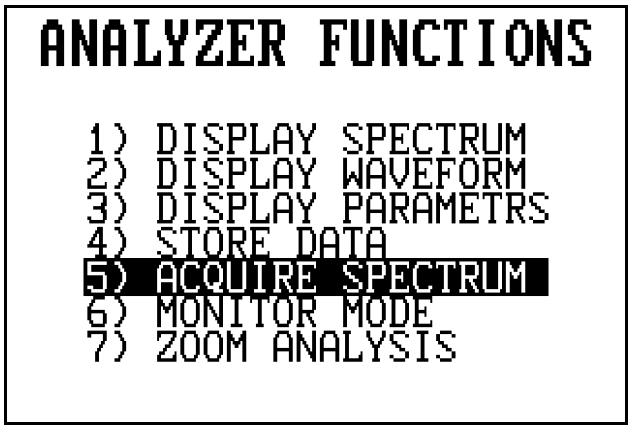
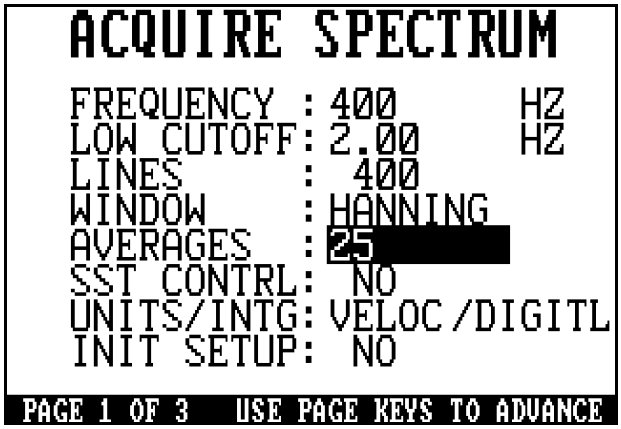
Press the **Analyze** function key to display the Analyzer Functions Menu. Highlight **5) ACQUIRE SPECTRUM** and press **Enter**.



Define the three-page menu. On page one, specify **25** averages.



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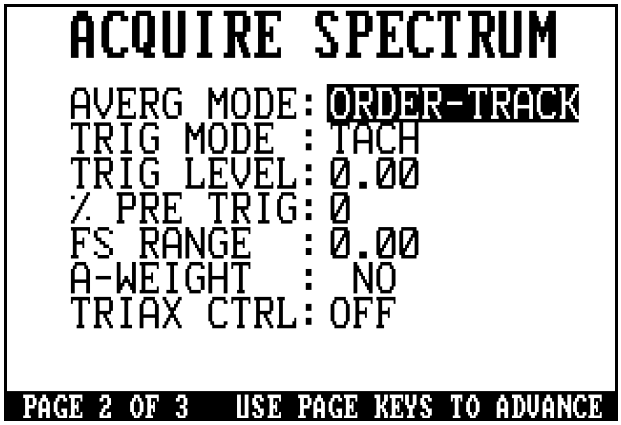
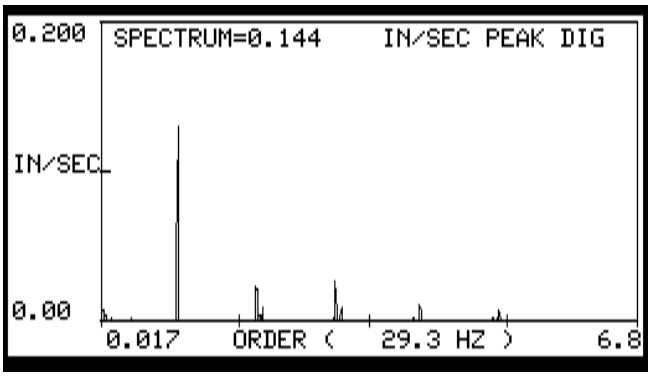
Toggle the Average Mode to **ORDER-TRACK**. Define the remainder of page two as shown below.



Press **Enter** to begin data collection. The spectrum was collected with **Order Tracking**. The average data displays with the frequency axis graduating in orders.



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Linear Minus Averaging

Linear minus (negative) averaging dynamically subtracts noise from an averaged spectrum. This method is one of the most powerful analysis tools available, because you can subtract data while the machinery is in operation.

Note: Noise is any vibratory energy undesirable for analysis. Signals that may be classified as noise include similar machinery vibration on a common base plate, belts, fans, and resonance.

The improvement of the signal-to-noise ratio gained by linear minus averaging is proportional to the square root of the number of averages performed. This improvement approximates -3dB for every number of averages equal to an even power of 2. For example:

- Measurement 1: Averages = $2^n = 2$ (n = 1)
- Measurement 2: Averages = $2^n = 4$ (n = 2)
- Measurement 3: Averages = $2^n = 16$ (n = 4)

Measurement 2 results in -6dB improvement in the signal-to-noise ratio.
Measurement 3 results in -9dB improvement in the signal-to-noise ratio.

Negative Averaging is a three-step process.

Step 1	Step 2	Step 3
Machine Running	Machine Running	Machine Running
Impact Machine	No Impact	No Impact
Normal Averaging	Negative Averaging	Step 1 - Step 2
Spectrum Displays: Normal Vibration + Result of Impact	Spectrum Displays: Normal Vibration	Spectrum Displays: Result of Impact

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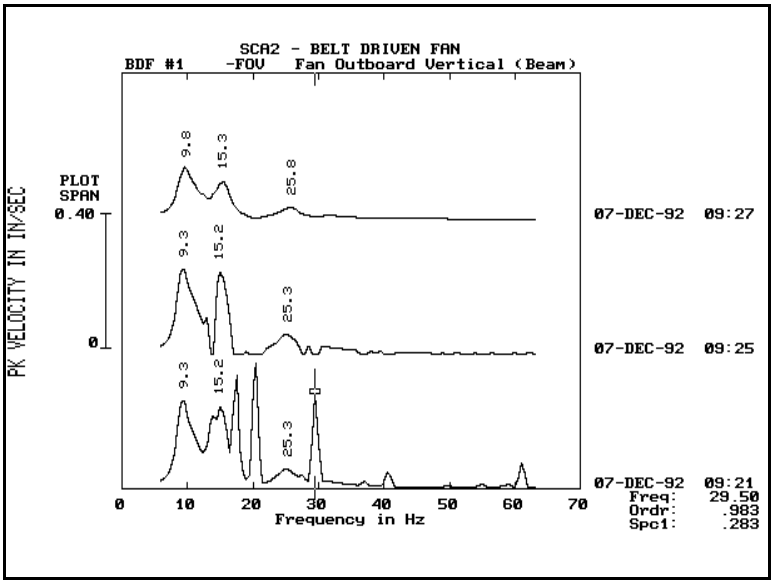
Negative Averaging Case History

The case history provides an example of linear minus averaging. The data comes from a belt driven fan assembly involved in a dynamic impact/resonance test. The multiple spectral plot below shows data taken from the same machine at different periods of time. The top plot shows impact data collected with the machinery off. The middle plot shows data collected with the linear minus function. The bottom plot shows data collected with the normal averaging mode.



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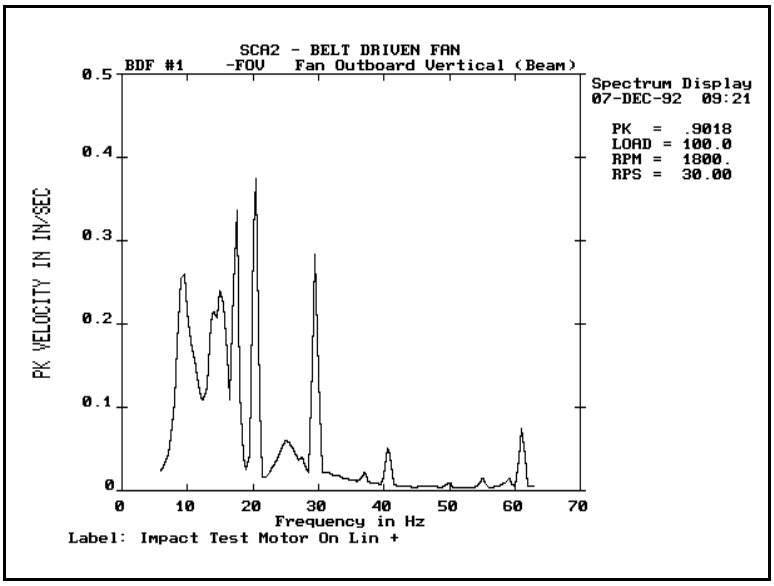
As discussed earlier, the ability of the linear minus function depends on the time given for noise subtraction. The ultimate objective of this function is to subtract any vibration data from the plot. If given enough averages, the unwanted data draws down to the noise floor of the analyzer (10^{-16}).

The support data shown in the next three plots appears in the order the analyst gathered it. The analyst collected data during the machine's normal run condition. He used normal averaging while impacting the structure approximately every two seconds. The analyst used this data for the dynamic impact information. After he collected enough dynamic impact data, the analyst ended the normal averaging mode and stored the data.



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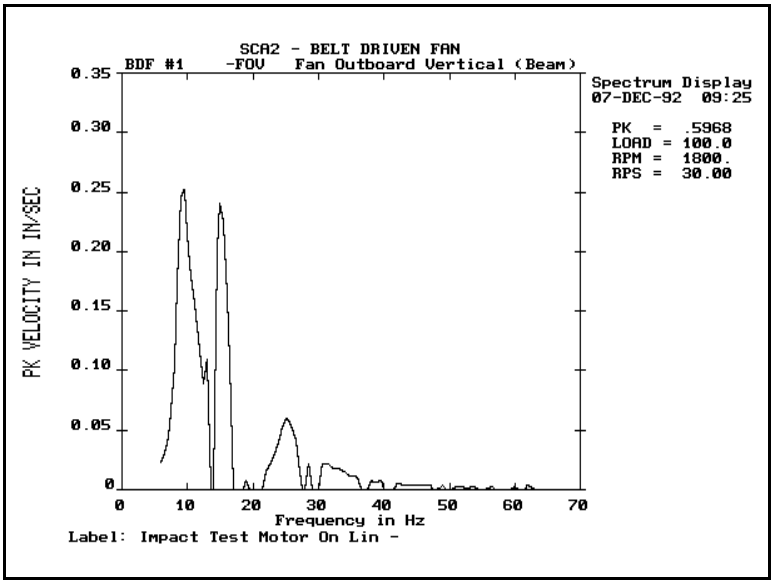
At this point, the analyst changed the averaging mode from normal (linear plus) to linear minus.

Note: The machine's operating condition was not interrupted. Also, note the absence of the operating peaks located in the previous plot. The analyst subtracted them from the data plot above. This function left only that data directly caused by the impacts.



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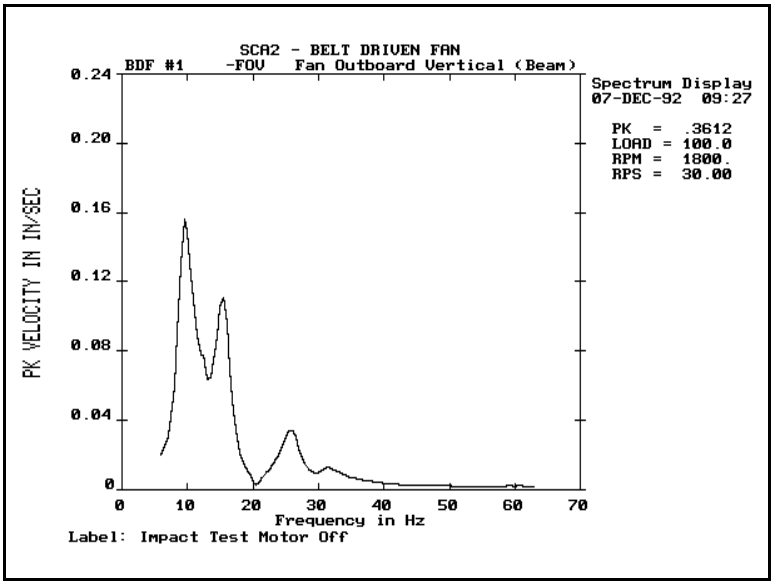
The data below is the impact on the machinery in the off condition. Note that this plot and the preceding plot are almost identical. The only differences are the holes left from the linear minus function during operation.



This analysis technique lends itself to a wide range of machinery applications. This case history illustrates that the Negative Averaging technique proved that the machine was running near resonance and, when the machine was shut off, the resonant frequencies were confirmed.

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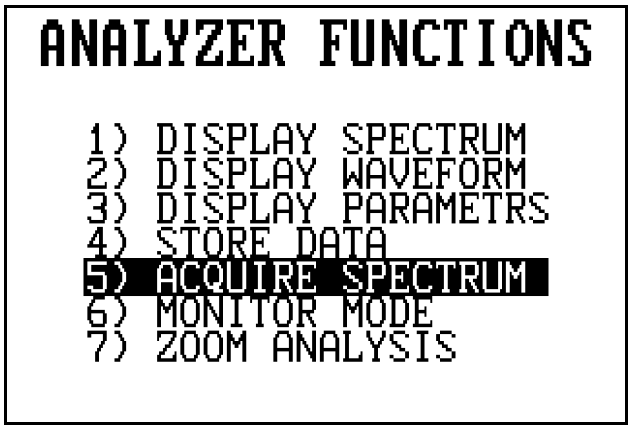
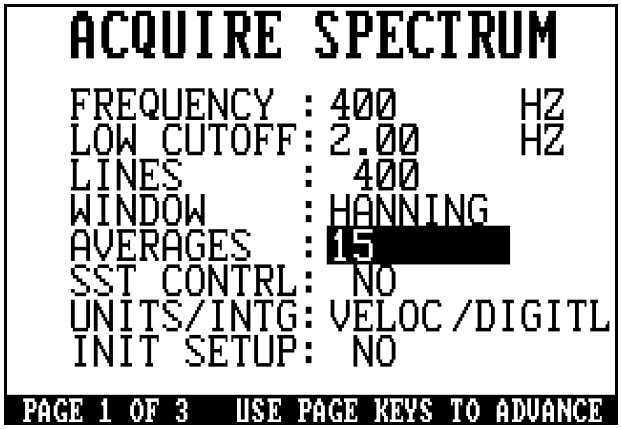
Press the **Analyze** function key to display the Analyzer Functions Menu. Highlight **5) ACQUIRE SPECTRUM** and press **Enter**.



Define the menus as shown. Negative Linear Averaging collects data first with normal averaging using the setup parameters given on the first page of the setup menu.



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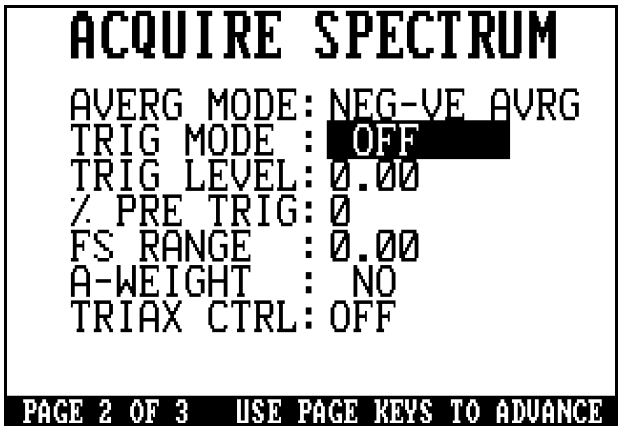
Press the **Page** key to advance to the second page of the menu. Toggle the Average Mode to **NEG-VE AVRG**. Press **Enter** to initiate data collection.



Press **Enter** again to start the data collection.

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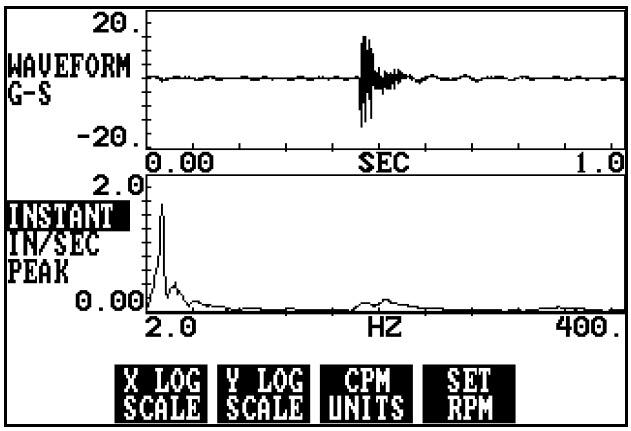
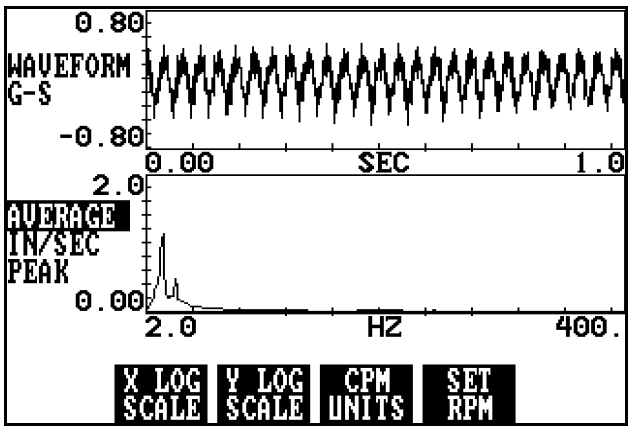
Begin impacting and press **Enter**. It is important to impact the structure while the analyzer is autoranging. This is a typical view of Step 1. Try not to get more than one impact per average. It is generally better to get no impacts in one average than to get two impacts in one average.



After collecting the data with normal averaging (with impacts), the analyzer displays a message. Press **Enter** to start the data acquisition with the linear minus averaging (without impacts).



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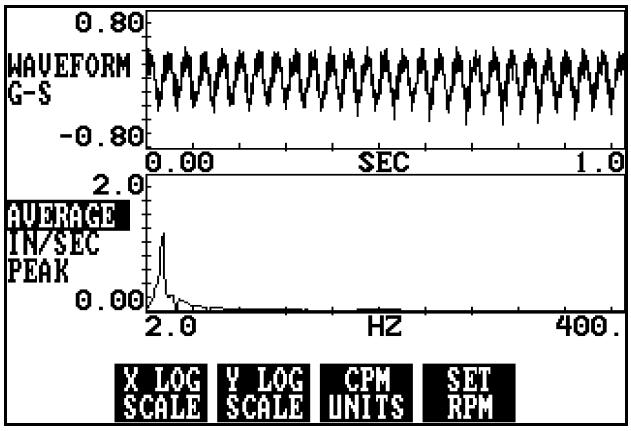
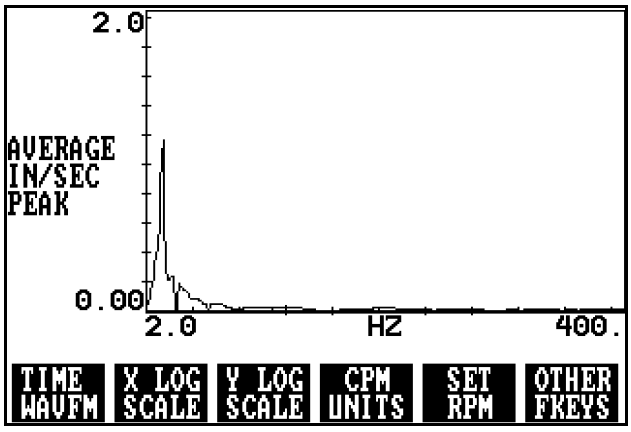
During Step 2, watch as the operational frequencies begin to decrease in amplitude.



After you are convinced that the operational frequencies have been sufficiently reduced, press **Enter** to stop the second step of data collection. The final step is the spectral displayed of the linear minus averaged spectra (data directly caused by the impacts). This spectra shows that a distinct resonance appears just below run speed. This data may be stored and displayed just like any other.



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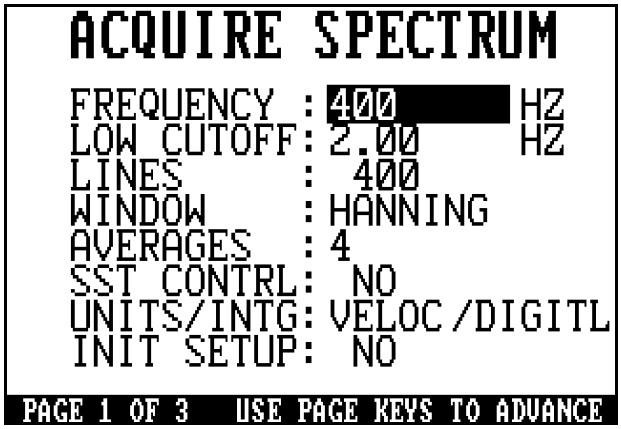
Data Collection Setups - Summary

The information below defines the functions of the Acquire Spectrum Menu. Each setup depends on the other. Therefore, it is important that you understand all the setup options.



FREQUENCY: — Specifies the maximum frequency, F_{max}, for data collection and display.

LOW CUTOFF: — Defines the lowest frequency displayed by the analyzer. Frequencies below this are still measured.

LINES: — 100, 200, 400, 800, 1600, 3200, 6400, 1/3 Octave.

WINDOW: — HANNING prevents leakage and is used most often; UNIFORM applies no shaping function and is often used for transient testing.

AVERAGES: — The number of samples to be collected. Selection depends on the Averaging Mode chosen.

SST CONTROL: — Activates a Low Frequency amplitude correction for Slow Speed applications.

UNITS/INTG: — Allows the setup of the spectral units and integration method without going to the **Utility** key and using the Change Setup option.

INIT SETUP: — Sets all the Acquire Spectrum choices to a response that matches the Route Point setup.

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The second page of the setup is not usually changed for most routine additional analysis, but you need to understand each of these choices.

```
ACQUIRE SPECTRUM

AVERG MODE: NORMAL
TRIG MODE :  OFF
TRIG LEVEL: 0.00
% PRE TRIG: 0
FS RANGE  : 0.00
A-WEIGHT  :  NO
TRIAX CTRL: OFF

PAGE 2 OF 3   USE PAGE KEYS TO ADVANCE
```

AVERG MODE: — Select between Normal, Peak Hold, Synchronous Time, Order-Track, and Negative Linear averaging modes.

TRIG MODE: — Select the type of trigger to be used for data collection. Select between Off, Normal, Tach, Pre-Trg, Pre-Tach.

TRIG LEVEL: — Units depend on trigger mode selected. Normal and Pre-Trigger modes require a vibration amplitude. Tach and Pre-Tach require an RPM value.

% PRE TRIG: — Percent of time waveform that will be collected before the trigger occurs.

FS RANGE: — Full scale range prevents autoranging, most useful for transient testing.

A-WEIGHT: — Frequency shaping method that approximates the response of the human ear. This is often used for airborne noise measurements.

TRIAX CTRL: — Can be used to select individual channels of a Triaxial accelerometer.

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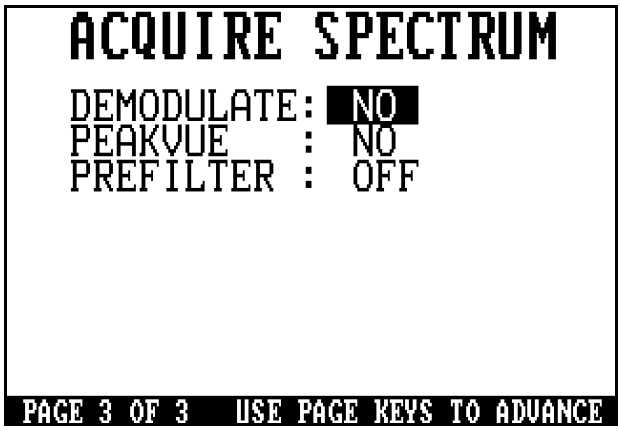
Page 3 of the acquire spectrum setup contains the DEMODULATE, PEAKVUE, and PREFILTER options.



DEMODULATE: — Set to YES activates demodulation of the signal using various high pass filters.

PEAKVUE: — Set to YES activates the PeakVue option, which allows the detection of very short duration, high-frequency impacts, which can be used to detect very early rolling element bearing defects and gearbox faults.

PREFILTER: — Allows the selection of various high pass, low pass, and band pass filters for both demodulation and PeakVue.

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PeakVue Filter Guidelines

50 Hz -- 5 KHz :	Felt problems, low to intermediate speed gearboxes
500 Hz --- 10 KHz :	Slow speed machinery (<900 rpm) including gearboxes with gear mesh frequencies <150 Hz
1.5 KHz --- 15 KHz :	Intermediate speed machinery (up to 1800 rpm) and gearboxes with gear mesh frequencies up to 400 Hz
5 KHz --- 40 KHz :	High speed machinery (> 1800 rpm) and gearboxes with gear mesh frequencies up to 2 KHz

Tip : To determine filter selections for gearboxes, select the filter whose lower corner frequency is greater than or equal to 3.25 times the largest gear mesh frequency.

Bandpass Filters (2120 Analyzer)

20 Hz ---- 150 Hz :	Felt problems on paper machines, etc:
50 Hz ---- 300 Hz :	Certain structural Resonance excitation, modulation of gear mesh in low speed machinery
100 Hz ---- 600 Hz :	Gear mesh modulation in intermediate speed machinery
500 Hz ---- 1 KHz :	Gear mesh modulation

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PeakVue Filter Guidelines

High Pass Filters

500 Hz : Low speed machinery having gear mesh <125Hz. Bearing and Gear problems

1 KHz : Intermediate speed machinery (<2000 rpm with gear mesh <300 Hz)

2 KHz : Medium speed machinery (<4000 rpm with gear mesh <600 Hz)

5 KHz : Machinery up to 9000 rpm and gear mesh to 1500 Hz. Requires attention to how the sensor is mounted as well as the sensor's frequency response.

10 KHz : High speed machinery with gear mesh up to 3000 Hz. Sensor "must"be permanently mounted with frequency response of 3 dB in the 30 KHz or higher range.

20 KHz : High speed machinery with gear mesh up to 6000 Hz. Sensor must be high frequency and permanently mounted.

Tip : Generally, use

Band Pass Filters when the event of interest is the excitation of a structural resonance, or the modulation of known frequencies, such as gear mesh.

High Pass Filters when the objective is to detect stress waves emitted by metal to metal impacting, fatigue cracking etc;

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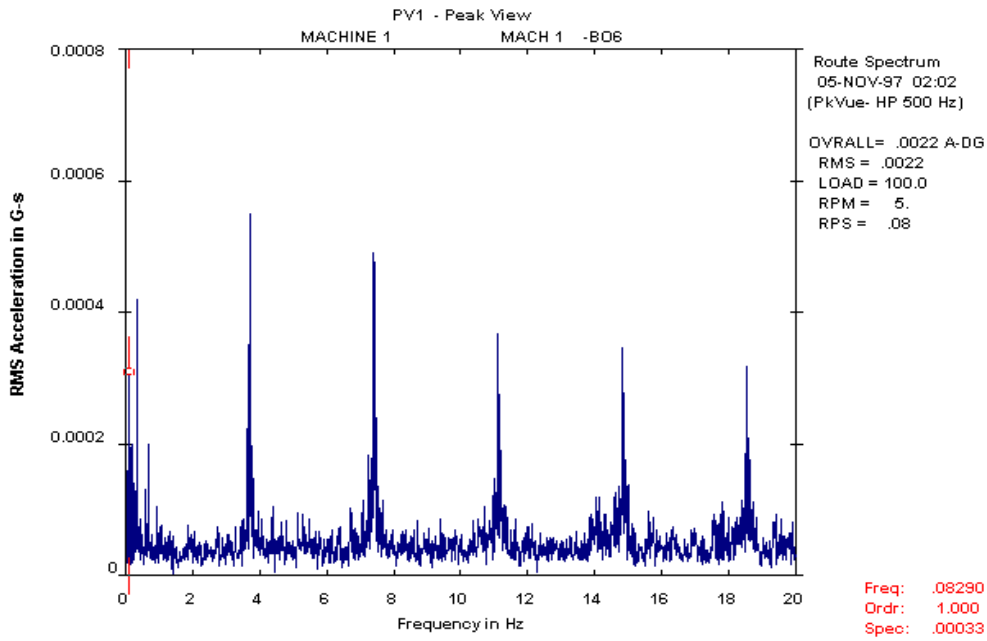
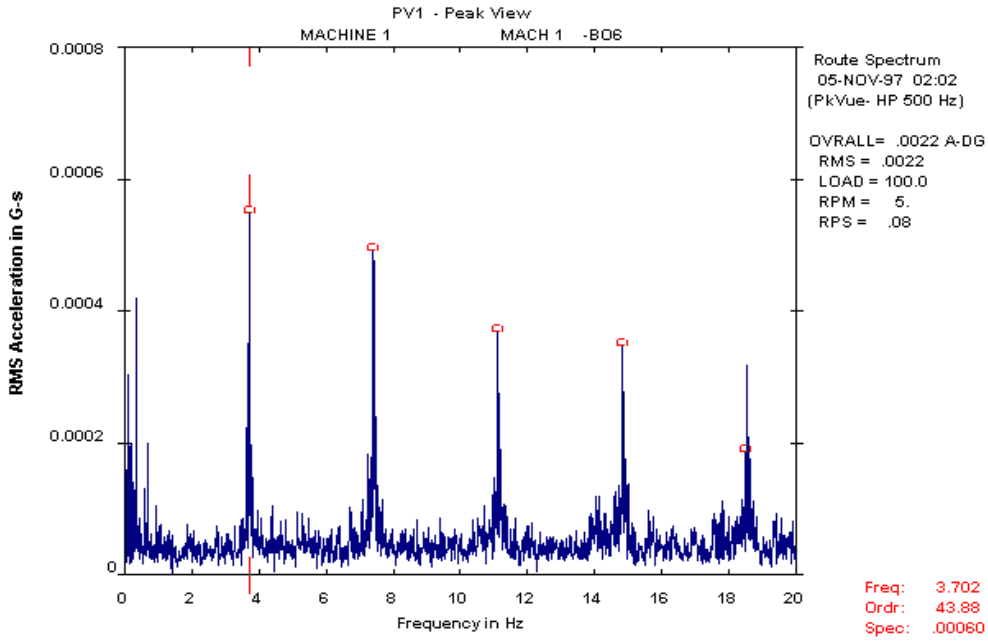
PeakVue Data



Notice, in plot above, the primary cursor is marking the turn speed at **5 rpm.**



In this plot displayed above the cursors are marking harmonics of the BPFO. We are viewing this data in acceleration. Notice the extremely low amplitude.

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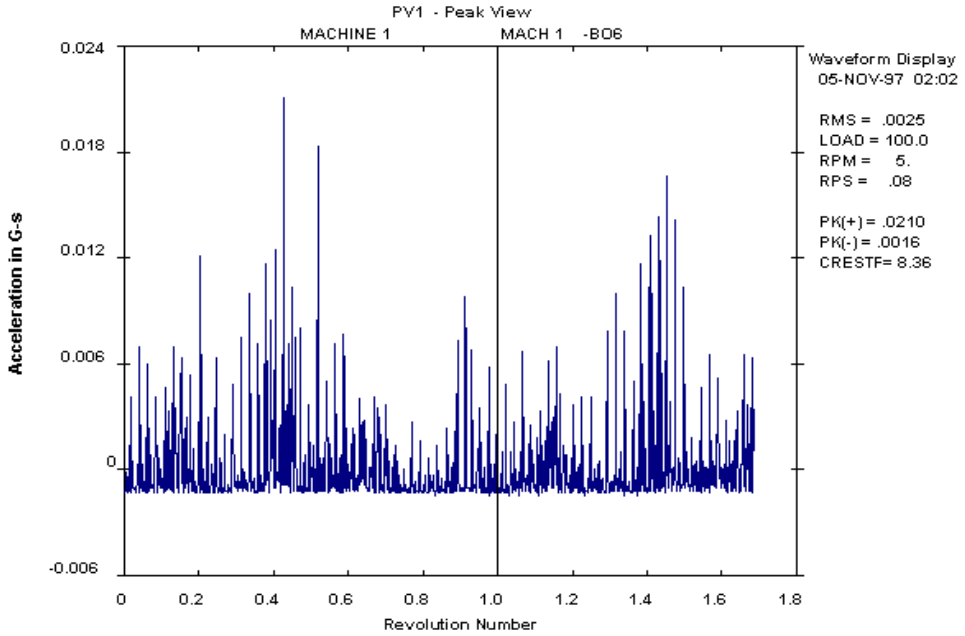
Displayed below is the waveform from the spectral data on previous page.

The Analyst must realize the data from PeakVue is reading **Stress Waves** that ordinarily would not be picked up by the transducer.

Also note that PeakVue rectifies the signal and places all the impacts on the positive side of the waveform.



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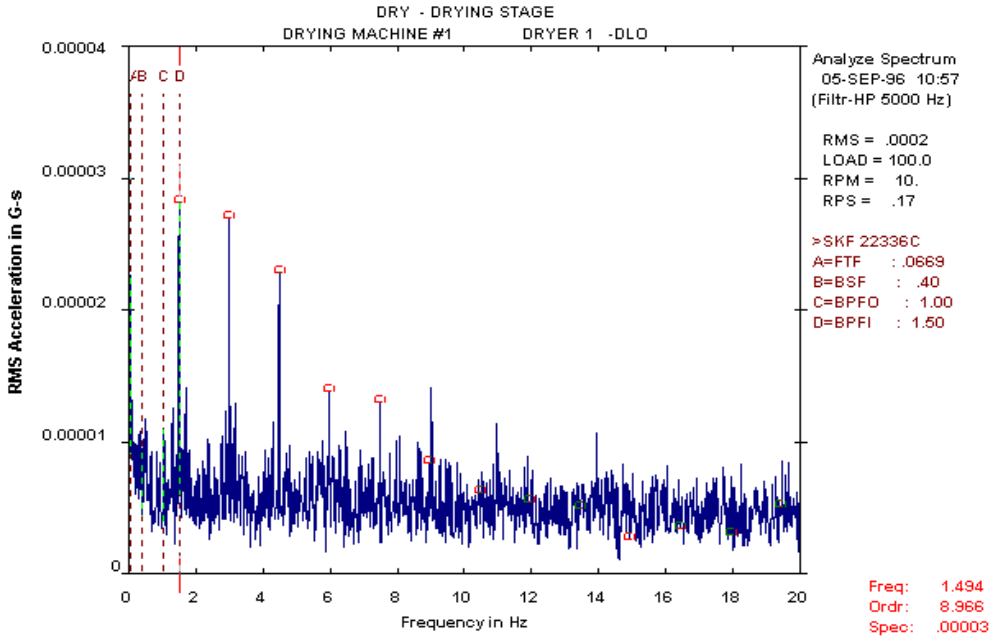
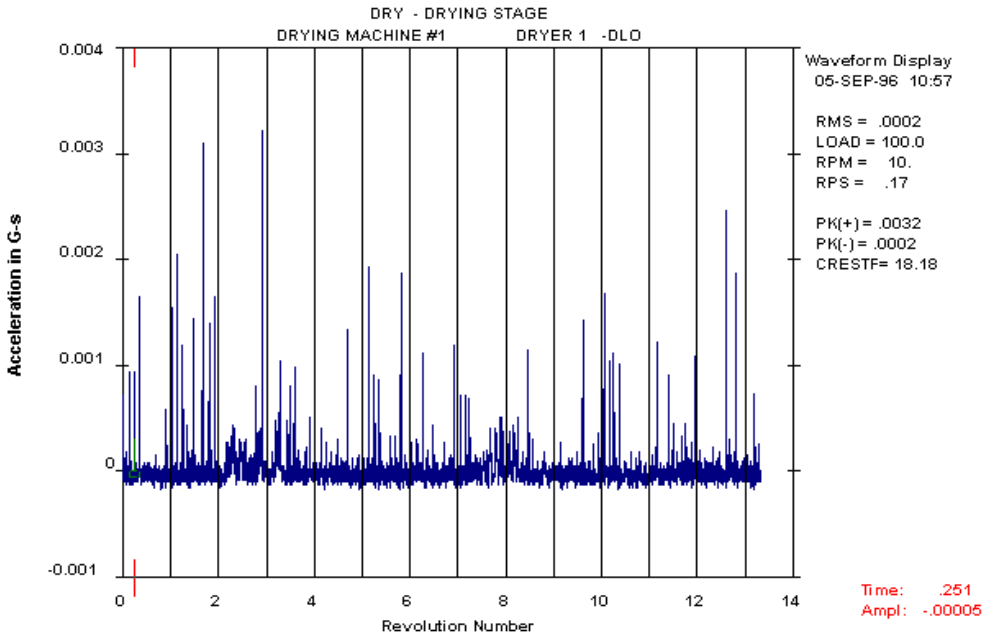
Displayed below is a plot of another machine diagnosed with a bearing problem using **PeakVue** technology.



Below is the corresponding waveform for the spectrum shown above.



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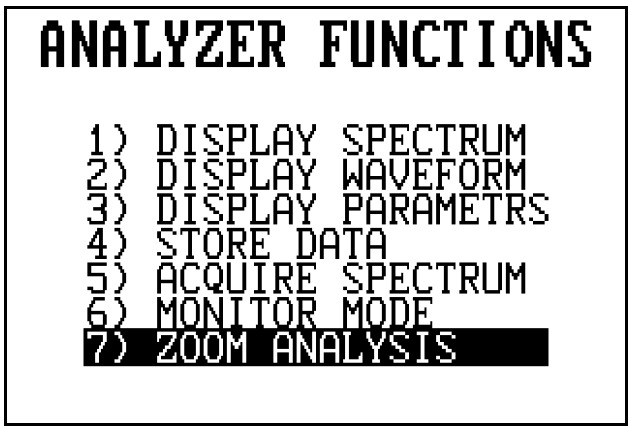
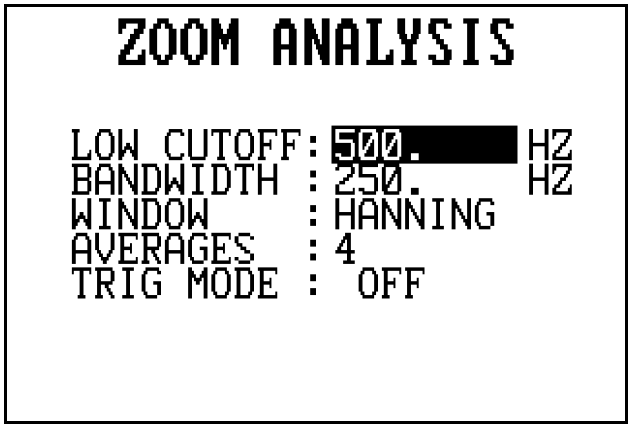
Zoom Analysis

Zoom Analysis can be used to place up to 800 lines of resolution between a lower frequency and an upper frequency for in-depth frequency analysis over a higher frequency range. Be aware of the fact that in some cases, better resolution (a smaller bandwidth) can still be achieved using the Acquire Spectrum option with 1,600, 3,200, or 6,400 lines of resolution. Select **7) ZOOM ANALYSIS** from the Analyzer Functions Menu, and press **Enter**.



Set the Low Cutoff to a frequency below your frequency range of interest. The Bandwidth defines the frequency range for data collection. Select the HANNING or UNIFORM window, the number of Averages, and the desired Trig Mode.



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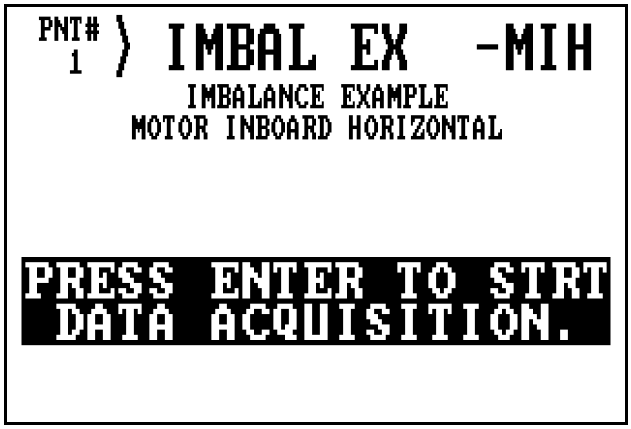
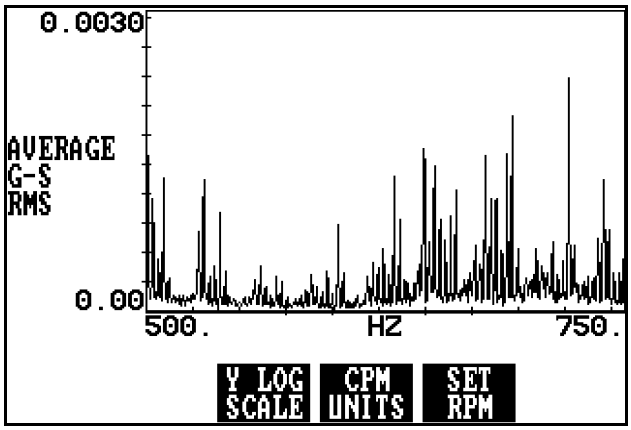
After pressing the **Enter** key, the screen below appears. Press **Enter** to begin data collection.



In this case 800 lines of resolution is being used over the frequency range of 500 Hz to 750 Hz. Depending on the preselected Low Cutoff and the Bandwidth, the analyzer toggles to the nearest internally predefined selections most closely matching your choices.



If the results of data collection are satisfactory press **Enter** and store the data. Only the spectrum is available for storage.

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"True" Zoom Analysis

The zoom feature in the 2120 provides what the industry has come to call "true zoom" capability. That is, the analyzer will collect a spectrum over a given frequency at a preset resolution. In the 2120, the preset resolution is 800 lines and you can move this 800 line "window" anywhere from 0 to 40,000 Hz (the f_{max} of the 2120).

This is different from our earlier analyzers in that the maximum resolution of the 2120 and 2115 is spread out over the f_{max} for each meter. For example, the most resolution you could have in a 2115 over 30,000 Hz is 3200 lines of 9.375 Hz/line (30,000 Hz/3200 lines). Although you could have MasterTrend collect a spectrum from 1800 to 2000 Hz to try to zoom in on this region, the total resolution is spread from 0 Hz to 2000 Hz. In this case with a 3200 line spectrum the resolution you would have in this 200 Hz range would be 0.625 Hz/line (2000 Hz/3200 lines) instead of 0.0625 Hz/line (200 Hz/3200 lines).

The zoom feature in the 2120 provides a moveable 800 line window. You can set the frequency range to anything you want. For example, you could monitor a gear frequency at 15,250 Hz by setting the zoom range from 15,150 to 15,350. This would give you 800 lines in this range or 0.25 Hz/line (200 Hz/800 lines). In the 2115 you could get 4.8 Hz/line (15350 Hz/ 3200 lines). Although you cannot collect zoom spectra in a regular route, you can save zoom spectra in analyze mode and dump it into MasterTrend with regular route data.

Here is a step by step process on setting up zoom in the 2120:

❶ Press the **analyze** key on the 2120
❷ Press **7** to go into zoom analysis
❸ Set the low cutoff below the frequency of interest
❹ Set the bandwidth to a frequency above what you are analyzing (this sets the frequency range).
❺ You may choose between **hanning** and **uniform** windows. Hanning is what is normally used.
❻ Set the number of averages for the spectrum. Four to six is standard.
❼ You may use a tach trigger or no trigger.
❽ To save your zoom spectrum to a route measurement point, press **enter** when prompted and then press **4** for store data. This will allow you to save the spectrum, the waveform or both. Note that you must have collected a normal route measurement for this feature to store your zoom spectrum.

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SST - Slow Speed Technology

The SST feature improves the quality of the very low-frequency vibration data generally encountered in slow turning machines. We will consider machines running between 300 and 600 RPM as slow speed. SST will make accurate measurements down to 0.1 Hz (6 RPM). A few important measurement considerations must be observed.

Use a low-frequency, low-noise, high-sensitivity accelerometer to collect data. Integrate the data from acceleration to velocity units using ANALOG integration.

Apply the SST correction feature to the measurement point. The SST feature corrects for the deterministic error occurring with the use of the analog integrator. The end result is the ability to see the low-frequency events at higher measured amplitudes, allowing for easier detection.

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Accelerometer Selection

To obtain the useful information required to perform analysis on slow speed equipment, a *low-frequency*, *low-noise* accelerometer will provide the results. The sensor should be minimally responsive to temperature measurement and should have a sensitivity of at least 500 mV/g.

Most accelerometers have a dynamic range of 100 to 120 dB. This means that the analyzer will have the limiting dynamic range. ***2120 has a range of 90 dB.***

If possible when choosing an accelerometer, a ceramic piezoelectric crystal is preferable to a quartz crystal and a shear mode accelerometer is preferred to a compression mode accelerometer.

When comparing displacement, velocity and acceleration, it is evident that displaying the data in units of displacement enhances the low-frequency data and acceleration depresses the low-frequency data. However, a drawback is that a displacement probe must be permanently and securely mounted so the portability factor is lost.

Placing a displacement probe on each measurement point also increases the equipment cost for your program. Integrating the data to velocity may be the best compromise.

When integrating data, it is best to use ANALOG integration. As mentioned in the previous section, there are a number of reasons for this choice.

❶ Analog integration attenuates the vibration signal above the F_{max} of the spectrum and thus improves the dynamic range of the analyzer in the lower frequency region.

❷ Analog integration reduces the low-frequency flare-up known as ski slope, which digital integration can actually increase.

❸ Analog integration produces a known effect (deterministic error) on the vibration data that the Model 2120 Machinery Analyzer can correct with the SST (Slow Speed Technology) feature.

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Practical Considerations

- Handheld mounts are unacceptable, because you cannot hold the sensor steady enough.

- From a cost perspective, you recover the money spent for a better accelerometer through the hours of time saved during data collection. Using a marginal accelerometer requires extended averaging to improve the repeatability of the data. This approach always yields measurements that take more time and incur more errors.

- Consider the vibration environment of the machine and its supporting structure. Low-frequency vibrations are not attenuated by structures and cannot be attenuated by any practical scheme of mechanical shock absorbers or dampeners. If a machine has an internal vibration of 5 mils P-P with a structural vibration of 10 mils P-P in the same low-frequency band, you cannot conduct vibration analysis.

- In general, special equipment and procedures required for monitoring extremely low-frequencies do not intermix easily with regular PDM data collection. Successful programs handle low-frequency monitoring in separate routes with analysis parameter sets (APS) tailored for each measurement point. Tailoring APS minimizes the collection time for each point without corrupting the integrity of the data.

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2120 Setup

This feature is accessible from the 2120 Acquire Spectrum Menu. Be sure and follow the integration rules discussed earlier. Press the **Analyze** key at the top of the 2120.

```
ACQUIRE SPECTRUM

FREQUENCY  : 40        HZ
LOW CUTOFF : 0.00      HZ
LINES      :  800
WINDOW     : HANNING
AVERAGES   : 4
SST CONTRL : YES
UNITS/INTG : VELOC/ANALOG
INIT SETUP :  NO

PAGE 1 OF 3   USE PAGE KEYS TO ADVANCE
```

Press the **Page** key to examine the setup for the other three pages.

```
ACQUIRE SPECTRUM

AVERG MODE : NORMAL
TRIG MODE  :  OFF
TRIG LEVEL : 0.00
% PRE TRIG : 0
FS RANGE   : 0.00
A-WEIGHT   :  NO
TRIAX CTRL : OFF
ACTIVE CHN :   A

PAGE 2 OF 3   USE PAGE KEYS TO ADVANCE
```

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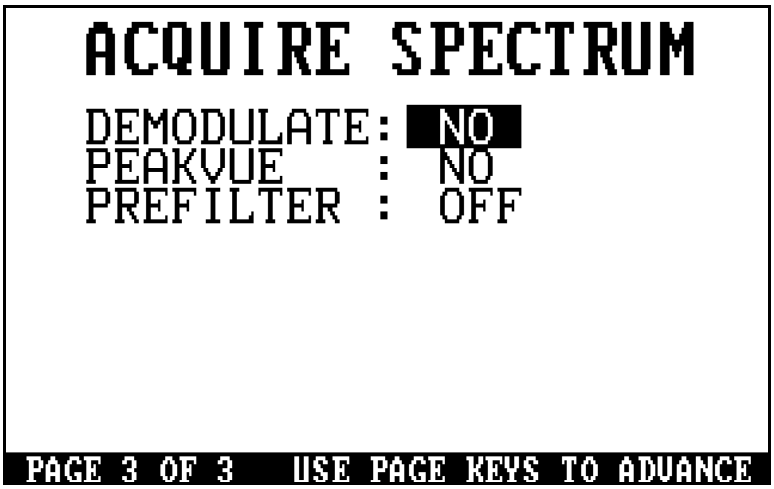
Generally speaking the Demodulation and PeakVue Methods should not be used at the same time as the SST feature, so these can be left with the **NO** response.



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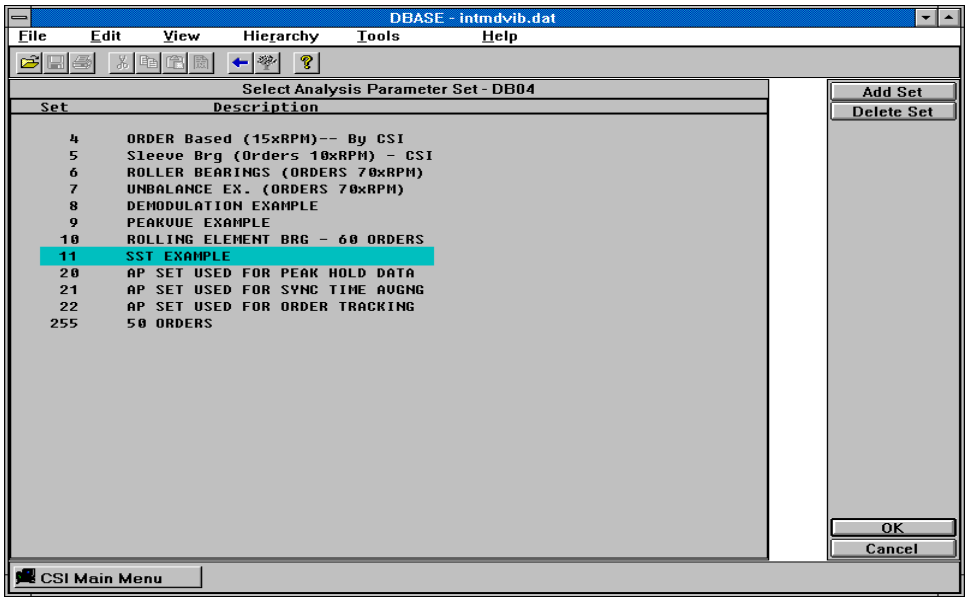
MasterTrend Setup

Again, this technique can also be selected from MasterTrend. Let's examine the analysis parameter set information needed for this application.



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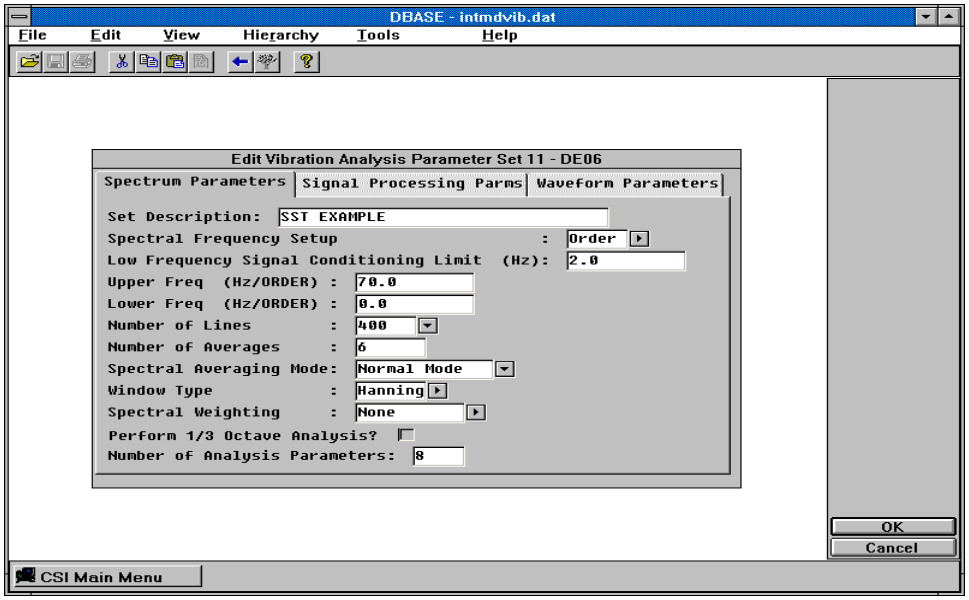
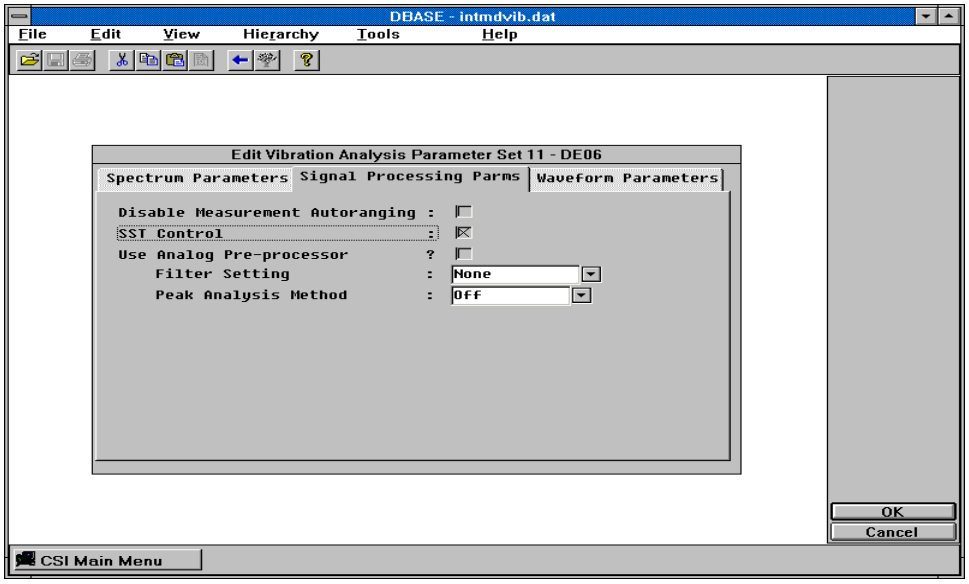
We have a point setup with standard parameters. Remember, at slow speeds (less than 600 RPM) the collection times will get long. You may consider lowering the LOR to 200. This should give an adequate bandwidth.



Under Signal Processing Parameters, select the SST Control feature. For most applications, the preprocessor should not be selected when the SST feature is used.



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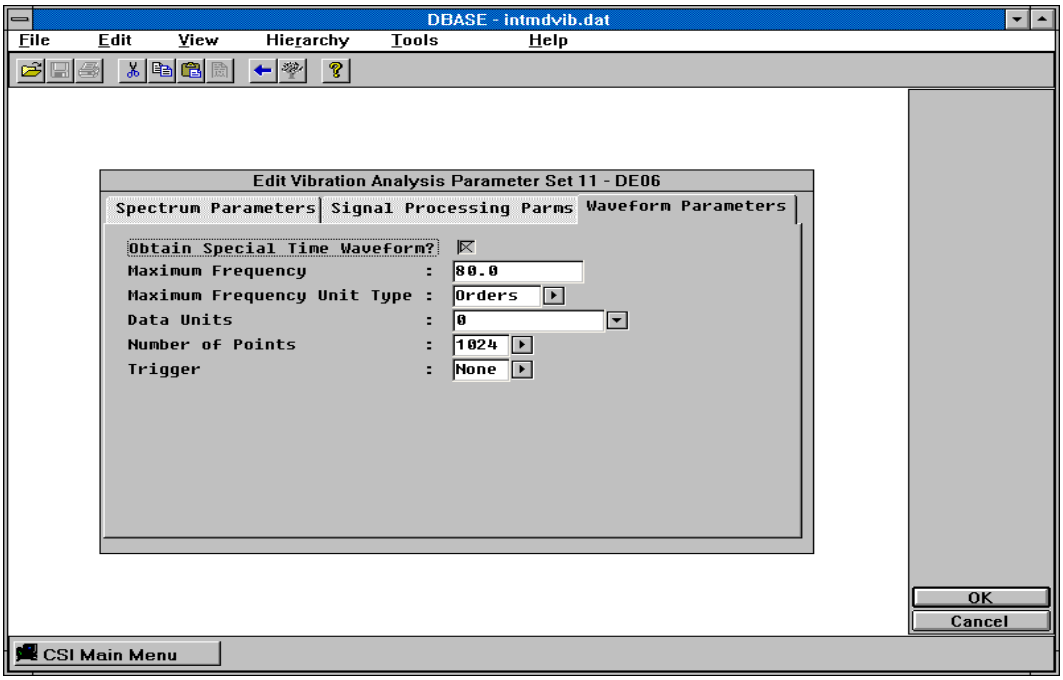
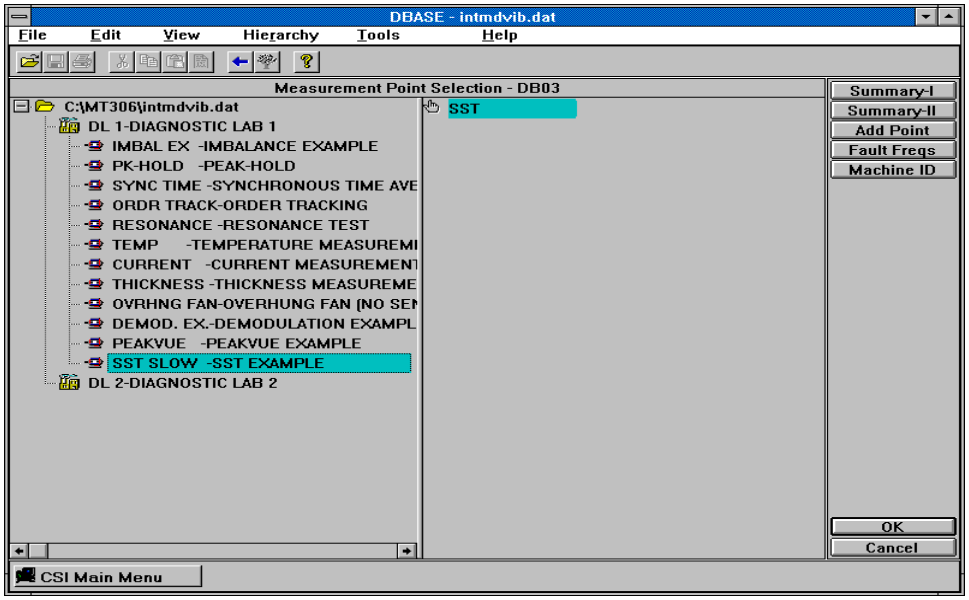
You may want to collect a special time waveform in G's acceleration. The SST feature uses analog integration which will put the waveform in velocity.



We also have a few measurement point setup considerations to look at. The **Units Type Code** should be **Vel <------ Acc**. We will use analog integration.



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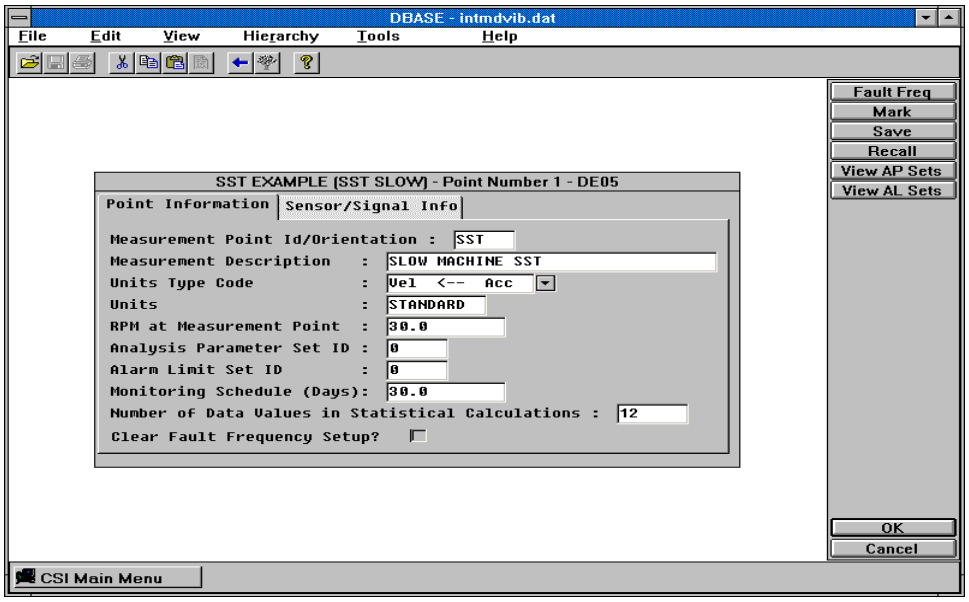
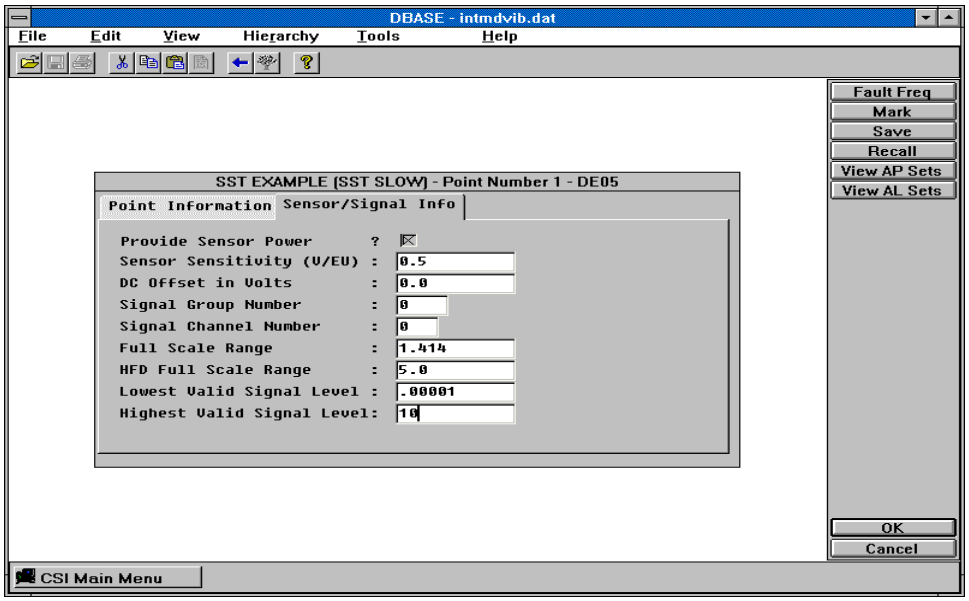


The second tab of measurement point setup will have the **Sensor Sensitivity** set to 0.5 V/EU or 500 millivolts per G. This is the standard slow speed accelerometer. Normal 0.1 V/G accelerometers should not be used. We will set the **Lowest Valid Signal Level** to **.00001**. This will take care of the expected low G level signals. Set the **Highest Valid Signal** to **10**. We should not see a signal larger than this.



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Low-Frequency Vibration Collection Lab

❶ Add a machine to your data base called SLOWONE.

❷ Add one measurement point. Remember we want to integrate the spectrum from acceleration to velocity. The sensor should be a 0.5 volts per g sensor.

❸ Create a parameter set to collect data using the SST method.

❹ Create a route using analog integration containing this machine.

❺ Download this route and collect data.

Summary

The information in this section provides not only a description of some of the choices available under the Analyze function key, but also some recommended selections for data collection. The features found in the Analyze function are very powerful analysis tools. Being able to use these tools will help solve many difficult and challenging vibration analysis problems. In later chapters, we will utilize some of the functions to aid in troubleshooting and analyzing various faults.

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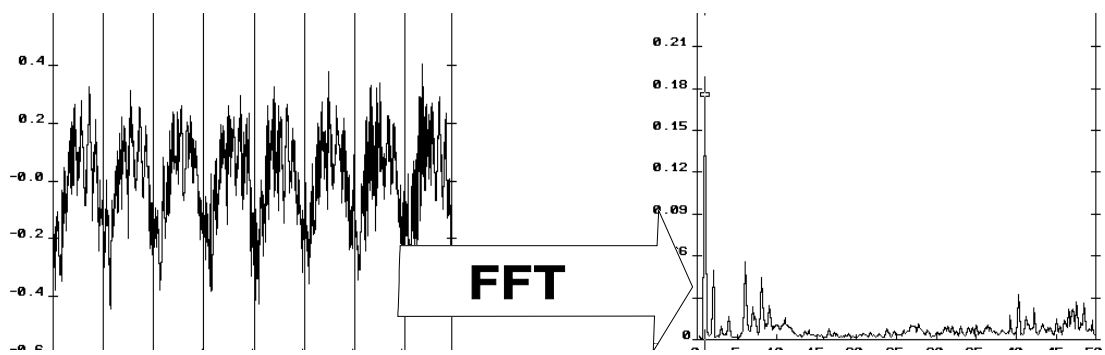
Section Nine
Digital Signal Processing

Objectives

- Relate time waveform length and frequency bandwidth to sampling rate and sample size.
- Choose the correct analysis window for each vibration analysis opportunity.
- Recognize limitations of digital signal processing.
- Demonstrate the ability to use the 2120 analyzer and the **Analyze** function to acquire data utilizing **Demodulation** and **PeakVue®**.

Fast Fourier Transform

The conversion of time domain information to frequency domain information is the ***Fast Fourier Transform*** (FFT).



Often a frequency spectrum is referred to as an FFT. However, the FFT is the mathematical conversion from the time domain to the frequency domain.

Since the signal that comes into the analyzer is an analog signal as discussed in the previous section, it must be digitally sampled by the analyzer. Therefore, the process used by digital analyzers is actually a variation of the FFT, called the ***Discrete Fourier Transform*** (DFT).

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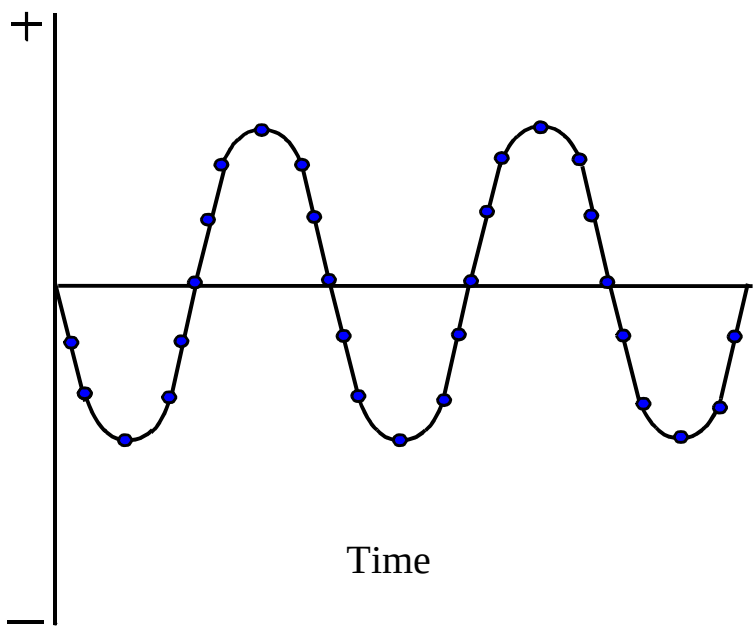
For the DFT, the time waveform is recreated in the analyzer by digitally sampling, and then it is transformed into the frequency domain.

The FFT process works based on the assumption that the signal measured and digitally sampled is one period of a periodic signal that extends to minus infinity and to plus infinity. Normally, this is true for most vibrating pieces of equipment.

It is the digital sampling process that makes the signal processing more complicated. The information here unlocks the mysteries of digital signal processing without getting bogged down in too much theory.



In order to understand the FFT digital sampling process, you must understand the relationship between lines of resolution (LOR), maximum frequency (F_{max}), length of time waveform (T_{max}), the digital sample size, aliasing, windowing, filters, and unit conversion.

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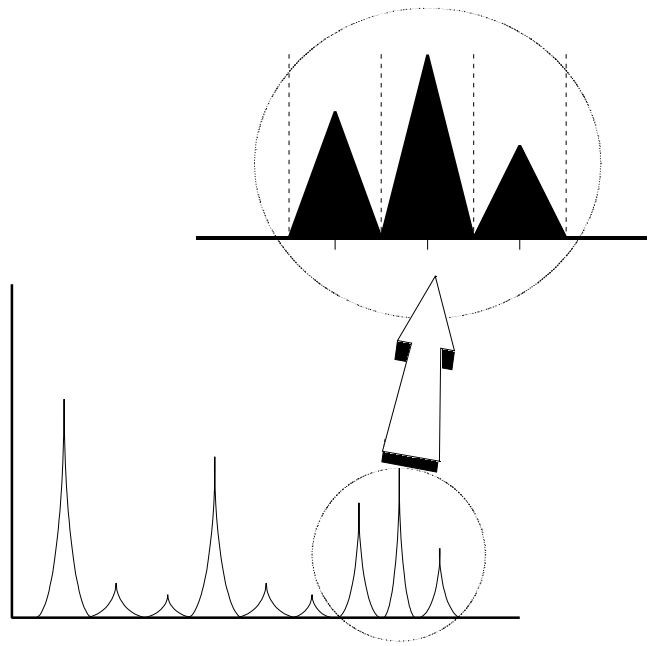
Resolution

Once data has been converted to the frequency domain from the time domain, view all the frequencies of interest in as much detail as possible. ***Resolution*** is the number of parts of the spectrum, usually called lines of resolution (LOR). The number of LOR selected are divided into the maximum analysis frequency (F_{max}) to arrive at the bandwidth (BW).

$$\mathbf{BW = F_{max}/LOR}$$



The lines are actually the center frequencies of *bins of energy*. Each bin contains an infinite number of frequencies, and all the energy in the bin is summed and represented by a single amplitude at the center frequency identified at each line of resolution.

First, identify the frequencies of interest so that enough resolution is chosen to separate closely spaced frequencies. A commonly used LOR is **800 lines**. Also, be aware that more LOR affect the length of the time waveform. For normal trending, we have to weigh the pros and cons of higher resolution. Remember that the time to collect one average is equal to 1 divided by the bandwidth. As the bandwidth decreases, the data collection time increases. The bandwidth (BW) should be no greater than 5 HZ/Line.

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This will give adequate resolution for identifying trend changes and reasonable data collection time. The time to collect data is dependent on the F_{max} and LOR settings as well as the Number of Averages and Data Overlap Percent.

The F_{max} setting is determined by the equipment being monitored as we covered in **Basic Vibration Analysis**. One popular way of setting F_{max} is to use an order-based set based on turning speed of the shaft being monitored.

Let's take a look at the effect of RPM on the BW with a typical 70 x Turn Speed Rolling Element Bearing Set.

RPM	**x 70 = F_{max}**	**÷ LOR**	**= BW (HZ)**
20 HZ (1200 CPM)	1,400	800	1.75
30 HZ (1800 CPM)	2,100	800	2.65
60 HZ (3600 CPM)	4,200	800	5.3

Since F_{max} ÷ LOR = BW and Time = 1 ÷ BW, we can calculate the time to collect data above with a typical 6 averages:

BW	**(1 ÷ BW) = $Time_{one\ sample}$**	**x # of Averages**	**= $Time_{total}$**
1.75	0.57 sec.	6	3.42
2.65	0.37 sec.	6	2.22
5.3	0.19 sec.	6	1.13

Actually, it is not quite this easy. We can increase our data quality with a Hanning Window and decrease our collection time with a technique called Overlap Averaging.

Sampling Time = [(1 - % of Overlap) (# Lines / F-max) (# of Averages)]

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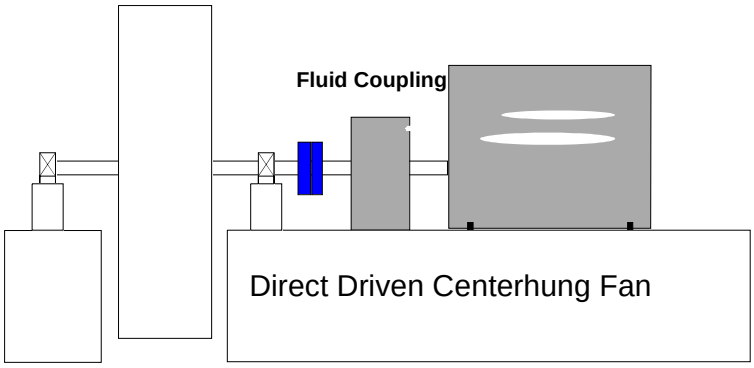
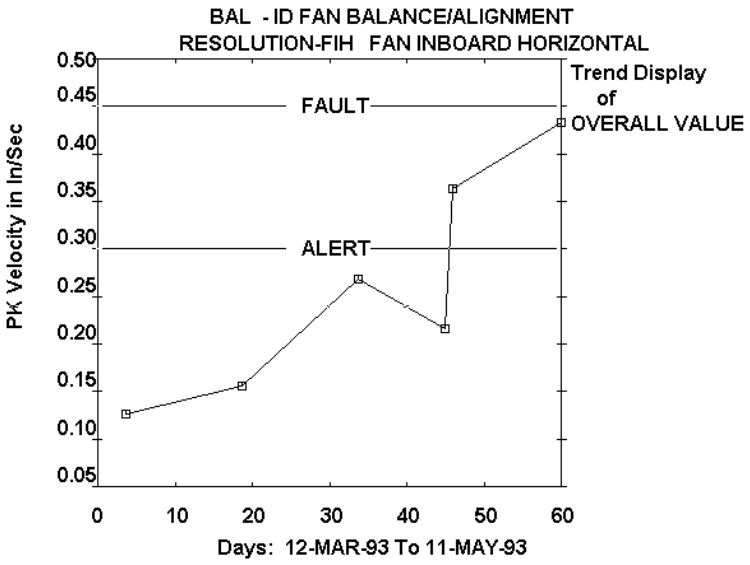
Case History- Spectral Resolution on I.D. Fan

2500 HP, 1175 RPM Motor



VA2CHSTY.DAT BAL-IMBALANCE, RESOLUTION-ID FAN BALANCE/ALIGNMENT

* 2500 HP, 1175 rpm Motor
* has a fluid coupling
* fan wheel 16 ft. in dia. and it has sleeve bearings .
* has a history of problems, most of the vibration on the FIH point
* fan moves 1200 degree air in a large kiln in cement plant



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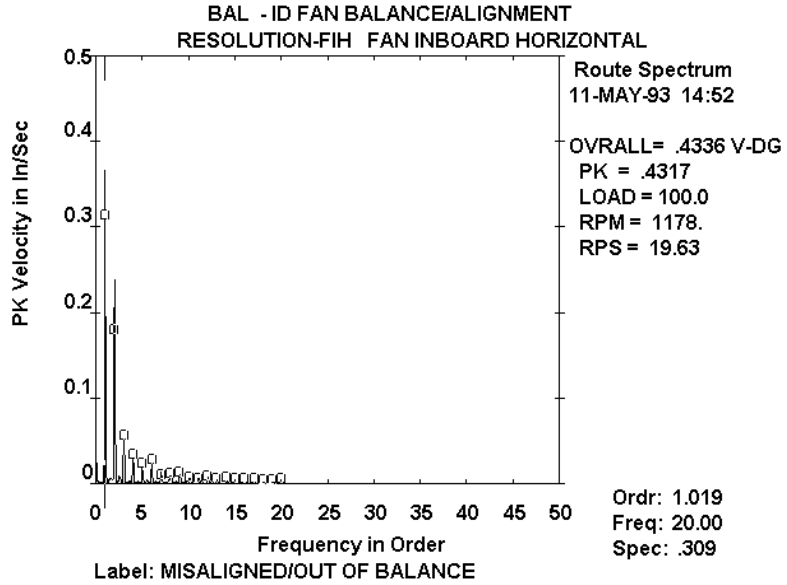

The peak at 1X RPM is high with 2X also high.

The fan was subject to buildup and abrasive wear, and imbalance was cited as the most likely cause with looseness secondary.

All the bolts were checked for correct torque and base inspected with no abnormalities noted. The maintenance manager made plans for the Vibration Analyst to return that evening to balance the fan.

The analyst set up the analyzer with the downloadable balance program to take a reference run. While monitoring the peak-phase data, he noticed the phase was erratic.

The amplitude data was swinging from a low of 0.02 in/sec to as high as 0.4 in/sec! There are many different causes of peak-phase instabilities, of which closely spaced frequencies is one. This is known as a beat frequency. The closely spaced frequencies add to each other as the phase matches, and subtract as the phases separate.

The analyst decided to capture data with better resolution. The results are shown. The problem becomes clearer as we zoom in around 1X RPM to show two distinct frequencies.

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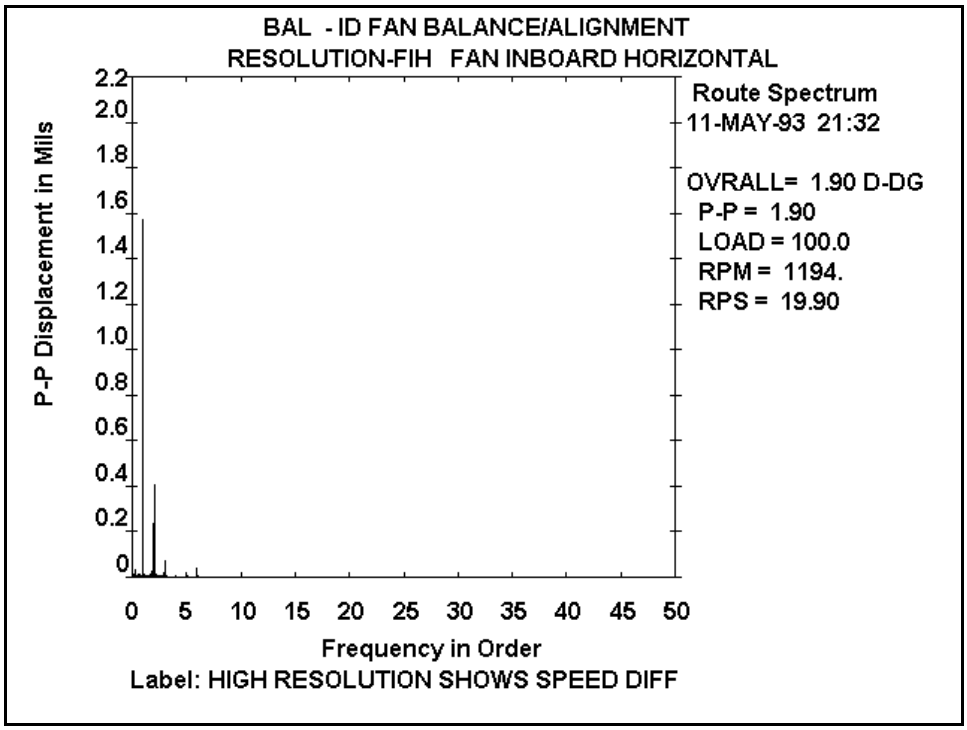

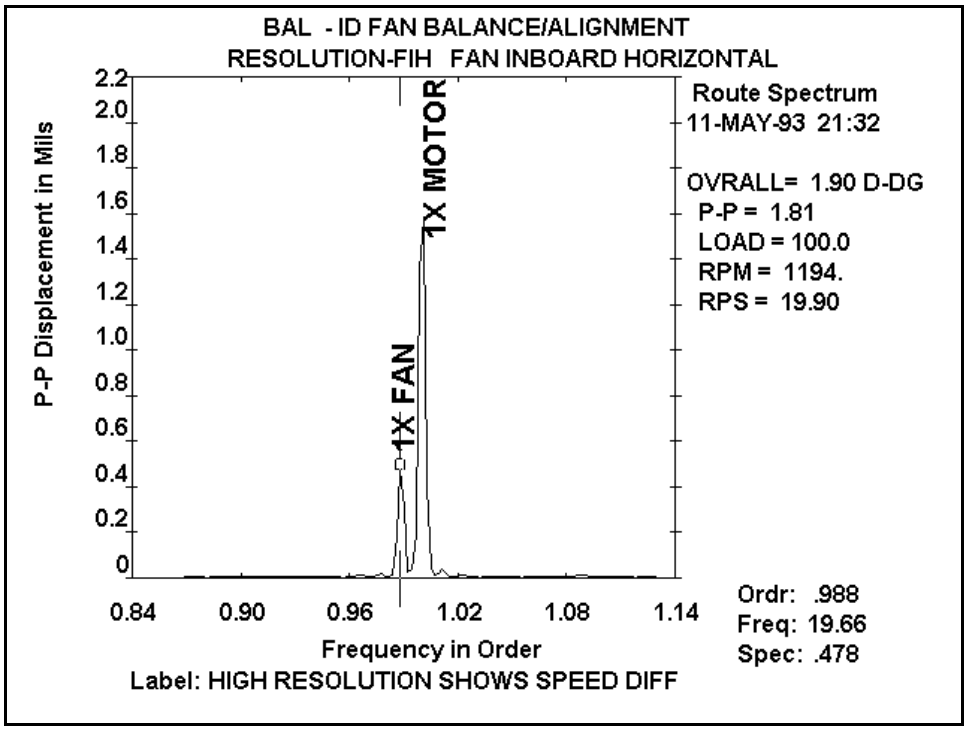

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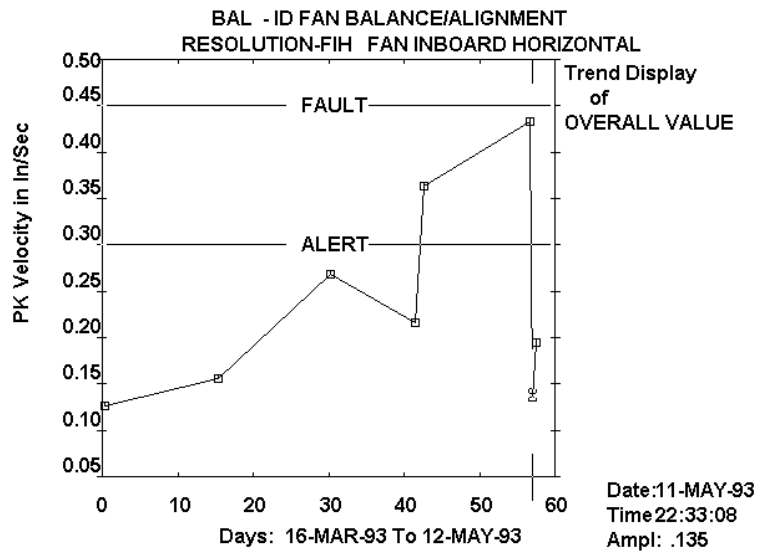
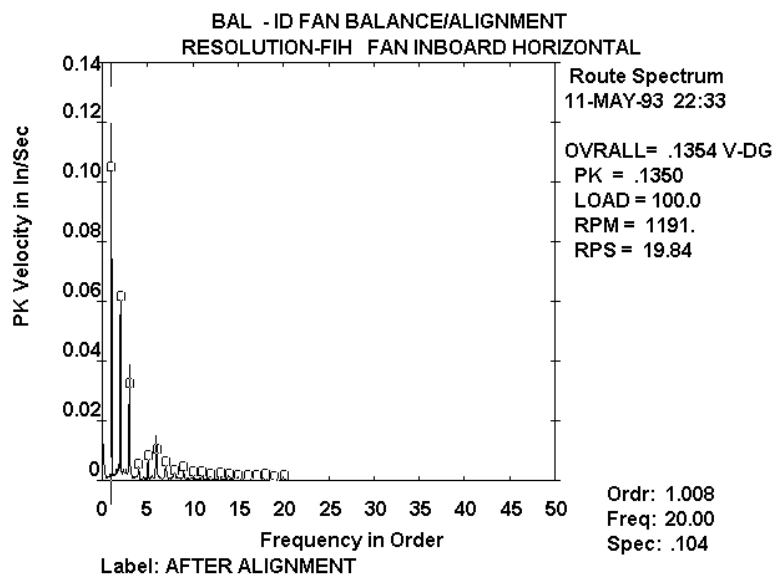
The fan vibration was 0.5 mils Pk-Pk. The motor vibration was 1.6 mils Pk-Pk. .***The fan did not need to be balanced.*** The high motor vibration on the fan bearings pointed to the fluid drive as the major contributor to high vibration. An alignment and fluid level change produced the following result.





Recognizing a beat frequency and utilizing the analyzer to troubleshoot and conduct on-site analysis allowed the maintenance personnel to quickly correct a problem. Vibration analysts have wasted many hours trying to balance equipment with beat frequencies with poor results. Bandwidth consideration is essential to quality route data collection as well as analysis/troubleshooting.

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Time Record Length

Calculate the time record length of the time waveform, T_{max}, from the following basic relationship.

$$T_{max} = 1 / BW$$

or

$$T_{max} = LOR/F_{max}$$

At face value, this is a simple and often used equation.

However, to understand the limitations of some analyzers, it is important to more fully investigate the relationship between the F_{max}, the LOR, and the T_{max}.

$$T_{max} = \text{Sample Size} / \text{Sample Rate}$$

$$\text{Sample Size} = 2.56 * \text{Lines of Resolution}$$

$$\text{Sample Rate} = 2.56 * F_{max}$$

These terms have already been defined, but be aware that some analyzers have an upper limit on the sample size.

In the 2120, the limit is 4,096.

* a 400-line spectrum would require 2.56 * 400 = 1,024 samples

* an 800-line spectrum would require 2.56 * 800 = 2,048 samples

* a 1600-line spectrum would require 2.56 * 1600= 4096 samples

* a 3200-line spectrum would require 2.56 * 3200= 8,192 samples.

However, if the analyzer is limited to 4,096 samples, then the 3200-line spectrum will be created from 4,096 samples since it is the upper limit of the analyzer.

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This is important when discussing the T_{max} in the time waveform, because:

- in general, raising the F_{max} decreases T_{max}
- and raising LOR increases T_{max} to the point that the multiple of 2.56 * LOR reaches the sample limit in the analyzer. In this case, the sample size for anything greater than 1600 lines is forced to be 4,096.

The table on the next page demonstrates how this limitation affects the T_{max} for a maximum 4,096 sample size analyzer.

$$T_{max} = \text{Sample Size / Sample Rate}$$

F_{max}	Sample Rate (S_r) = F_{max}*2.56	LOR	Sample Size (S_s) = LOR * 2.56	Time (sec.) = $S_s \div S_r$
400	1024	***400***	1024	1.00
400	1024	***800***	2,048	2.00
400	1024	***1600***	4,096	4.00
400	1024	***3200***	8,192 **4,096 stored by MT**	8.00 **4.0 stored by MT**

The last entry in the table may seem incorrect, but remember 4,096 is the maximum sample size, and anything that would be greater than 4,096 is forced to be 4,096 for storage in MasterTrend.

This is the reason your time waveform is not affected when following the raise-the-LOR-to-lengthen-the-time-waveform rule. You must be aware of the upper limit of the sample size and the number of LOR to which this number corresponds.

To increase the amount of time in the time record, it will become necessary to adjust the F_{max} to a lower value.

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The following chart shows the effect on the time record of various F_{max} settings.

F_{max} (Hz)	Sample Rate (S_r) = F_{max} * 2.56	LOR	Sample Size (S_s) = LOR * 2.56	Time (sec.)= $S_s \div S_r$
1000	2560.0	1600	4096	1.6
400	1024.0	1600	4096	4.0
200	512.0	1600	4096	8.0
100	256.0	1600	4096	16.0
10[1]	25.6	1600	4096	160.0[2]
10	25.6	3200	8192	320.0[3]

[1] ***Low Frequency Limit***
[2] ***Maximum Stored by MT***
[3] ***160 Stored by MT***

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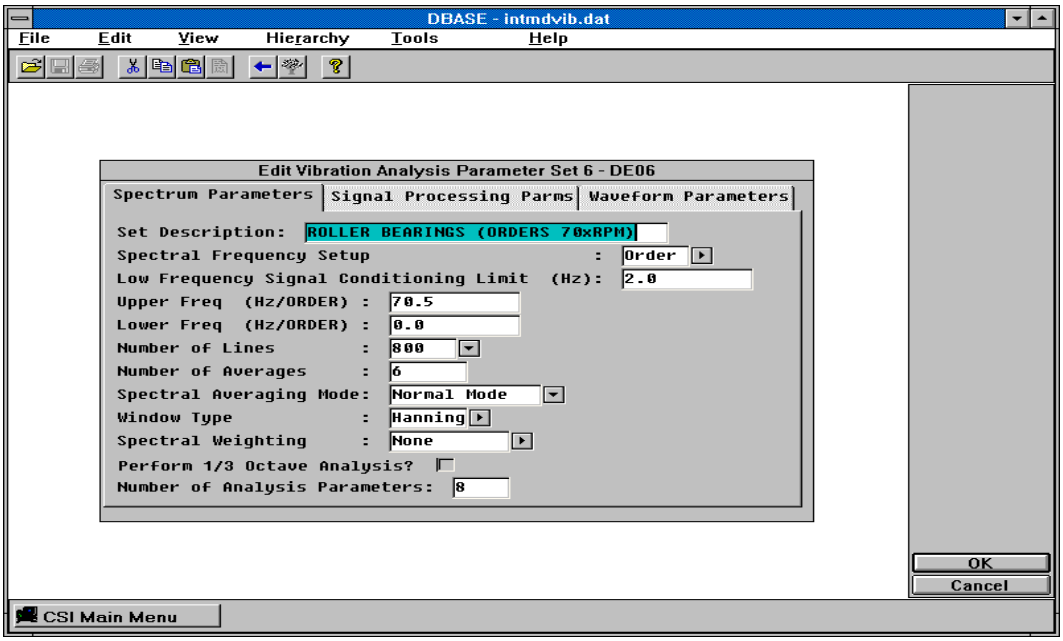
2120 Analyzer Lab - Setup-Time Waveform

Example: While collecting data on a 2-pole (synch. speed 3600 RPM) electric motor, the live time spectrum display shows a peak at 2X RPM that changes significantly in amplitude from average to average. In addition, an audible humming fluctuation is noted. Realizing this may be a beat frequency, you decide to view the time waveform for confirmation. With a typical analysis parameter setup of 70.5 X RPM and 800 lines of resolution, the time record is calculated as follows:



F_{max} * 2.56 = Sample Rate; LOR * 2.56 = Sample Size;

Time = Sample Size ÷ Sample Rate ***or*** Time = LOR ÷ F_{max}

F_{max} = 70.5 x 60 Hz (3600 RPM); F_{max} = 4500 Hz

LOR = 800

Time = 800 ÷ 4500 (2120 rounds to 5000 Hz)

Time = 400 ÷ 5000; Time = .16 seconds

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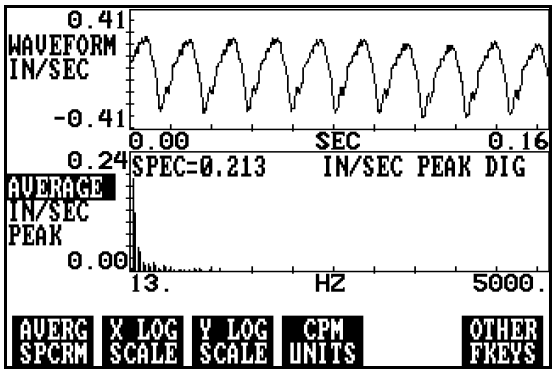
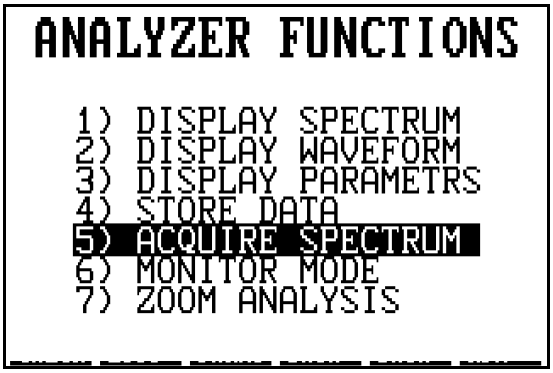
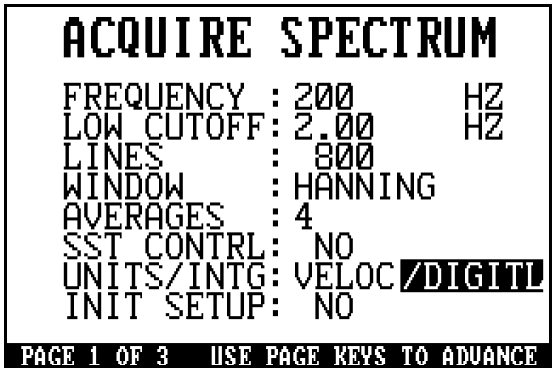


Obviously .16 sec is not enough time to see a slow beat frequency, so the Analyze function is very useful here. Let's say you decide that 4 to 5 seconds of data would be appropriate for this analysis. To obtain that data you can adjust F_{max} and LOR to meet your needs. A calculator will be helpful. A number of calculations will work — here is one:

Press the **Analyze** button and choose **ACQUIRE SPECTRUM**.





The **FREQUENCY: 200 HZ** (F_{max}) and **LINES: 800** (LOR) will give us the desired **4 seconds** of time record (400 ÷ 100 = 4).

This will also allow us to evaluate the spectrum around 60 and 120 Hz with high resolution. The Bandwidth will be calculated by: **200 Hz ÷ 800 Lines = .25 Hz/Line**

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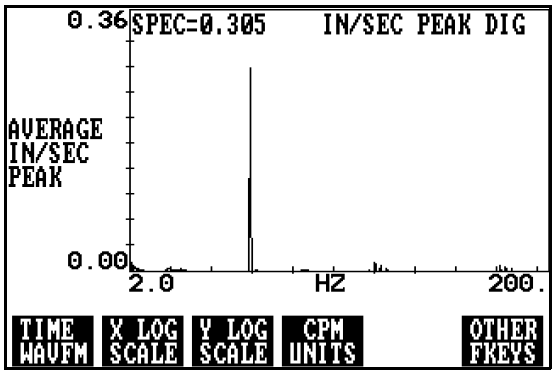
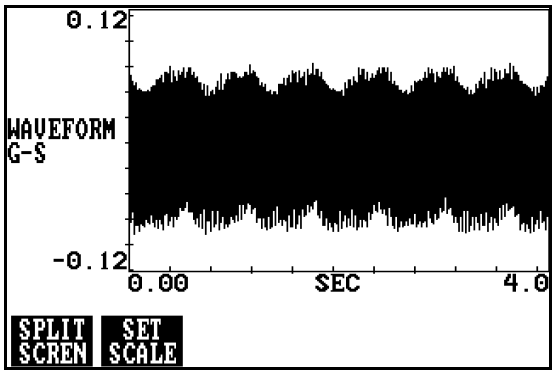
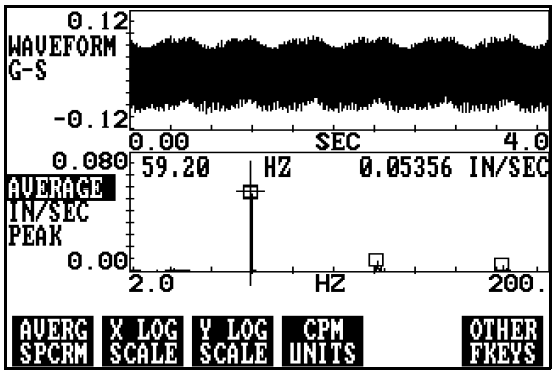
The resulting spectrum is shown below. Press **F1** for the time waveform.





Press the **F1 (SPLIT SCREEN)** key.

Notice the *spectrum* data (bottom) taken with an accelerometer has been **integrated** to units of **velocity**. The time waveform remains in units of **g's acceleration** because of the integration method chosen in the acquire spectrum. The choice of **UNITS/INTG: VELOC/DIGITL** will collect the data as shown.



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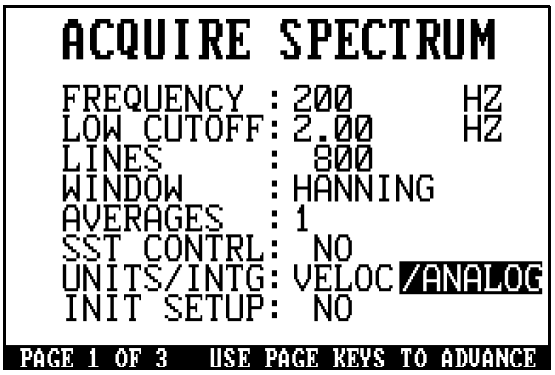
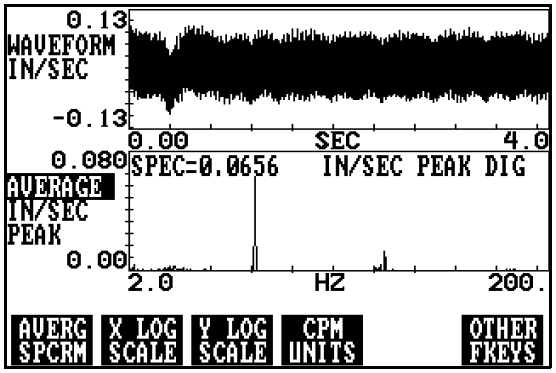
Now collect the data by changing the integration mode to **ANALOG** and view the results.

Notice the waveform data is in **IN/SEC Velocity**. Integration takes place in the time domain and therefore is integrated and stored in the integrated (velocity) units.





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When collecting a spectrum/waveform with the Analyze function of the 2120 for bearing and gear defect analysis, **digital integration** will be the preferred method.

This will store the waveform in acceleration units. Because of the relatively high frequencies generated, acceleration units makes it easier to determine impact severity as bearing and gear flaws develop.

Velocity has the effect of smoothing out impacting in the time waveform. In MasterTrend, we will use analog integration, but collect a **special time waveform in units of g's acceleration**.

Press the **Enter** key.

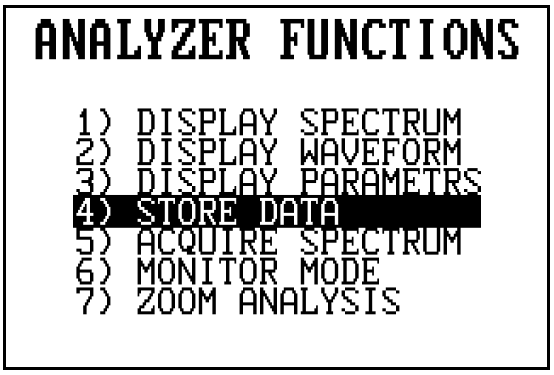
The **ANALYZER FUNCTIONS** screen appears.

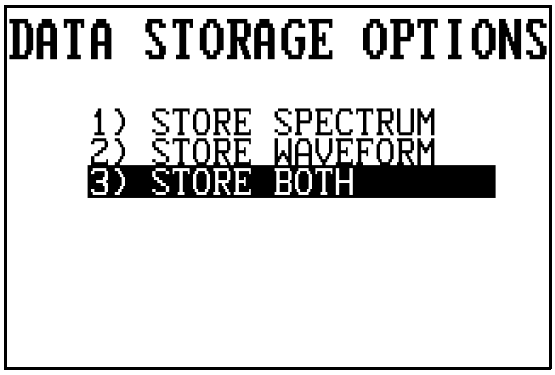
Select **4) STORE DATA** press **Enter**.

You have the storage option of spectrum, waveform or both.

Choose **3) STORE BOTH**.

This data will be attached to the current point as an **Analyze** data.





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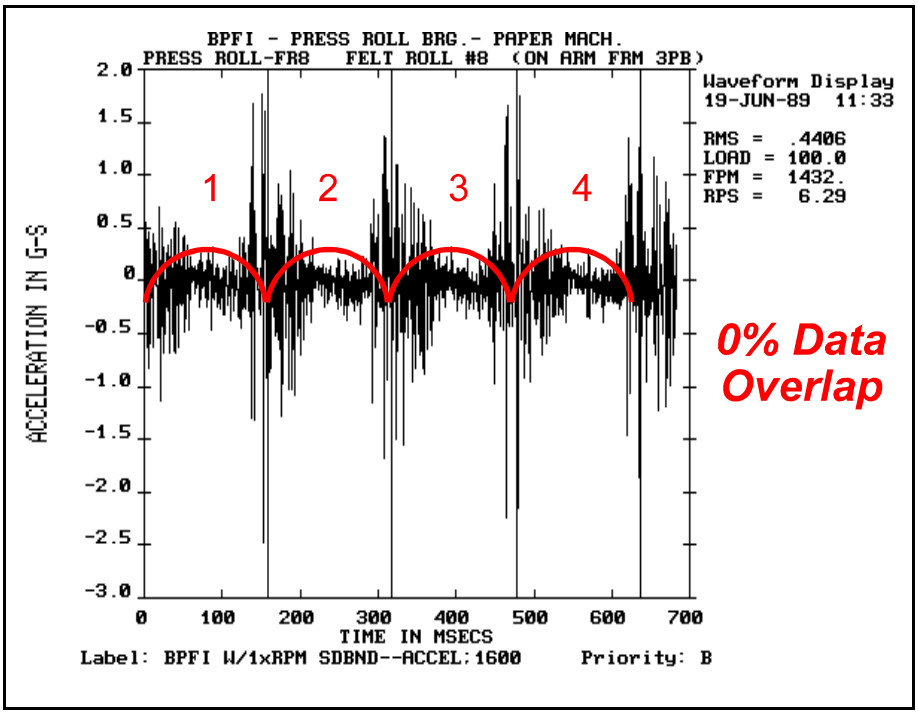
Overlap Averaging

Overlap averaging reuses data stored in the input time buffer of the analyzer.

When using a Hanning window on our vibration data, most of the data has been multiplied by a factor of one at the center of the window to zero at the beginning and the end of the time window.

In order to fully use all the data acquired in the analyzer, use overlap averaging. Fifty percent to 67 percent is recommended. Sixty-seven percent is the default



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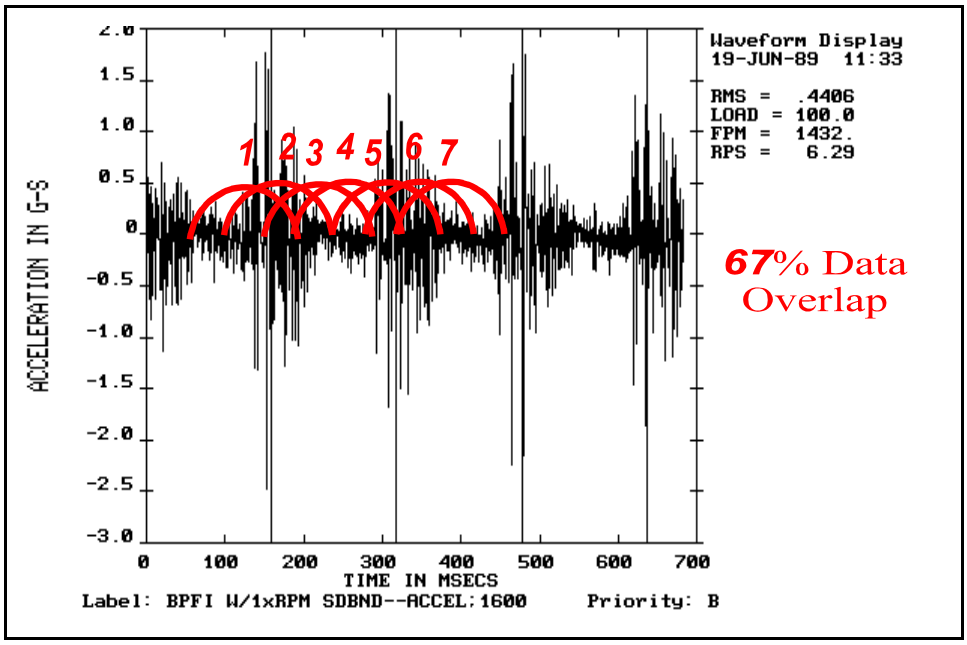
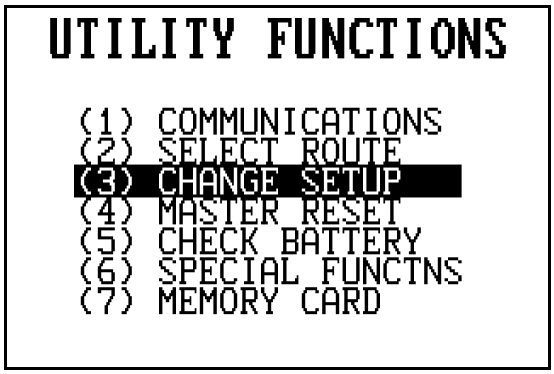
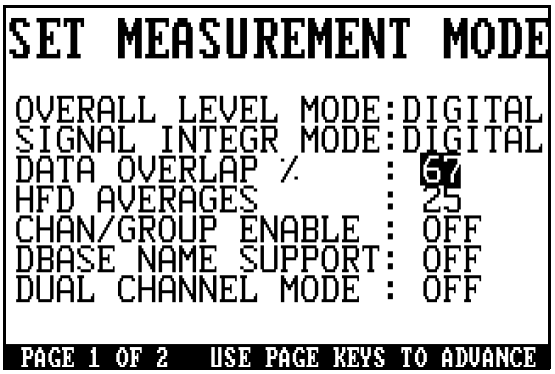


The default setting of 67 percent overlap averaging reuses 67 percent of the time data, saving acquisition time and generally improving amplitude accuracy. In the 2120 the setup is done with the utility function.





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Aliasing

The term ***aliasing*** brings to mind the idea of someone masquerading as someone else. Criminals often do this to elude law enforcement agencies. In the world of vibration analysis, an aliased frequency is one that appears as something it is not. Fortunately, it is not trying to hide from us but was forced into existence by an incorrect digital sampling process.

In a digital analyzer, the time domain data is digitally sampled at the sampling rate. We know that the sampling rate is controlled by the maximum analysis frequency (F_{max}).

In order to get good data, the sampling rate must be set higher than two times the maximum frequency of interest, F_{max}. Two times F_{max} is known as the ***Nyquist frequency.*** A very common sampling rate is set at **2.56 times the F_{max}.**

An aliased frequency occurs when frequencies higher than the F_{max} are present in the signal. The sampling rate under-samples this high frequency and creates a low (aliased) frequency equal to the high frequency minus sampling frequency.

For example:

If the F_{max} is 1,000 Hz, then the sampling frequency will be:

1,000 * 2.56 = 2,560.

If a frequency is in the data at 2,768 Hz, then an aliased frequency can be expected in the data at 2,768 to 2,560 = 208 Hz.

To avoid aliasing, digital analyzers use ***anti-aliasing filters*** that remove all frequencies above 40% of the sampling rate before the time data is converted to frequency data.

Therefore, with most digital analyzers, aliasing is no longer a concern.

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Leakage

Leakage occurs when complete cycles of the vibration signal in the time waveform are not captured in the time record.

Instead of a discrete frequency in the spectrum, energy spills into adjacent bands or lines of resolution. A rectangular window can allow time data to be captured that cannot be transformed into the frequency domain. If a non-integer multiple of the periods of vibration appears in the time waveform for all the frequencies present, the frequencies smear over many lines of resolution.

Digital analyzers use windowing functions to avoid smearing. Some typical window choices are:

Uniform - which is really no window at all

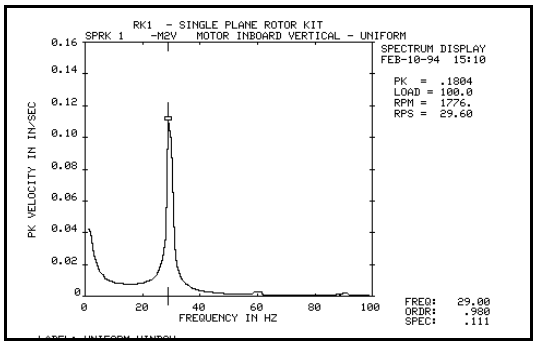
Hanning - (often called a cosine taper)

Flat Top, **Kaiser Bessel**, **Force**, and **Exponential**.

The purpose of windowing data is to eliminate leakage problems. For most vibration analysis applications, **the window of choice is the Hanning window**. The Hanning window does a great job of forcing the beginning and the end of the time record to a zero amplitude. This makes the reconstructed time waveform continuous with no amplitude variations.

A sinewave acquired with a uniform window that has an integer number of periods in the waveform transforms into the frequency spectrum with all the energy contained in one spectral line of resolution. However, this condition is difficult to achieve.

If the time waveform contains a non-integer multiple of periods, then the frequency data smears over many lines of resolution.



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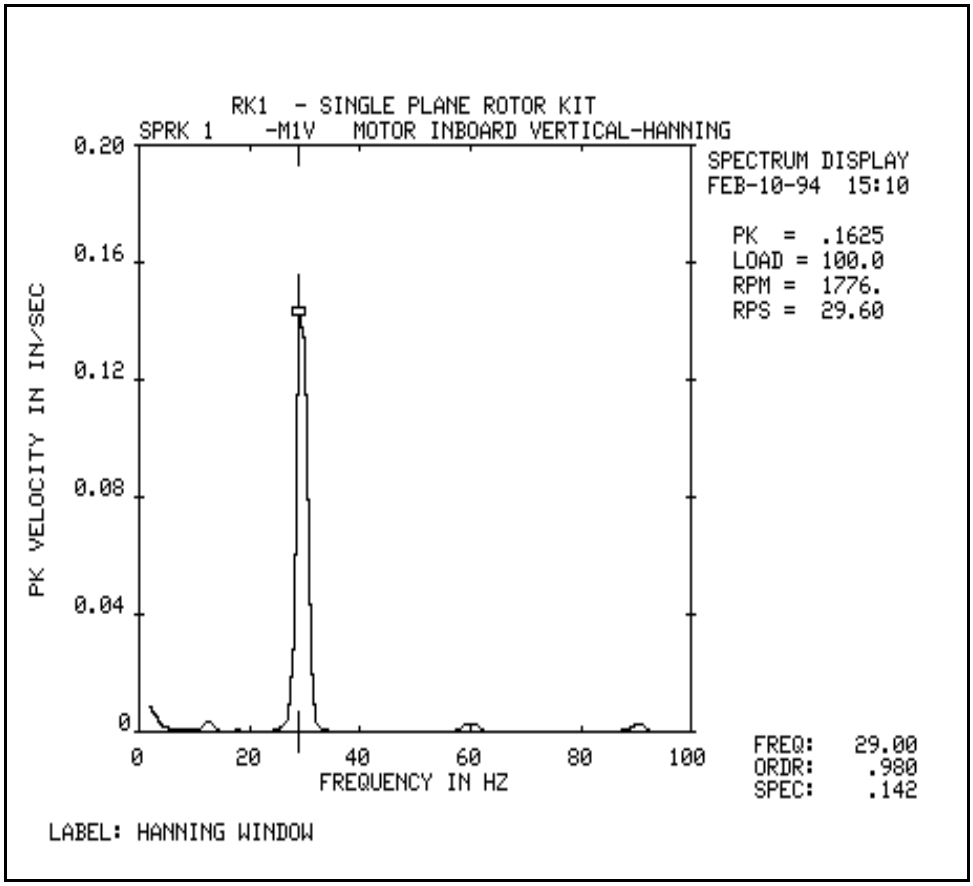
The most typical case involves using a ***Hanning*** window on a non-integer multiple of periods in the time waveform. In this case, the data transformed to the frequency domain is contained primarily in one line of resolution with some frequency data smeared into the adjacent bands of the primary frequency. This, however, tends to be the most acceptable method of data collection.



The Hanning window still spreads a pure discrete tone over three lines of resolution. To separate very closely spaced frequencies, increase the lines of resolution a little more.

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Integration and Differentiation

The vibration data that is the input signal into the analyzer is a time-varying voltage proportional to the vibration measured by the transducer. In other words, an accelerometer produces a voltage that varies over time relative to the acceleration measured by the transducer. The voltage amplitude in the time waveform is converted to the desired amplitude units based on the sensitivity and conversion factor of the transducer.

Most analyzers have the ability to convert from the measurement units of the transducer to either of the other two units in the time domain or the frequency domain.

At CSI, integration of the time signal is called ***analog*** integration and integration of the frequency domain is called ***digital*** integration.

Integration is a process of converting from acceleration to velocity or displacement, or converting from velocity to displacement.

Differentiation is the process of converting from displacement to velocity or acceleration, or converting from velocity to acceleration.

How are these unit types related, mathematically?

D (Displacement) = distance traveled by vibrating object
V (Velocity) = change in Displacement/change in Time
A (Acceleration) = change in Velocity/change in Time

These terms are often represented with the following equations:

$$D = X$$
$$V = X / T$$
$$A = X / T / T = V / T$$

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Therefore, if any one of these terms has been measured, integration and differentiation allows any of the other terms to be calculated, provided the analyzer or software used is capable of this conversion process.

One drawback to **integration** is a flare-up of the lower frequency data caused by the integration process. This effect is often called ***integration noise*** or a ***ski-slope effect***.

This is typically only noticeable when integrating from acceleration to displacement and tends to affect only the lower one percent of the frequencies. This may cause the overall vibration level to be higher than usual, and it may be included in the calculation of the overall vibration level.

Normally, this is not of any real concern unless the analyst wants frequency analysis below 2 to 4 Hz. If this is the case, the use of **SST** may be beneficial.

Filters

The vibration industry generally uses three types of filters:

- **Low Pass**
- **Band Pass**
- **High Pass**

Each one filters data out of a signal, which you may find useful when analyzing signals with large dynamic ranges.

For example, some spectra have both large and small amplitudes relative to each other. Because of the dynamic range of the analyzer, however, you cannot analyze the low amplitude vibration in the same plot as the high amplitude vibration.

A filter helps resolve the problem.

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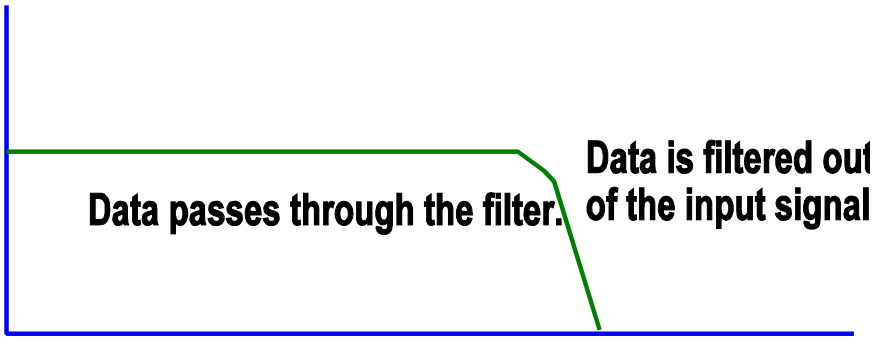
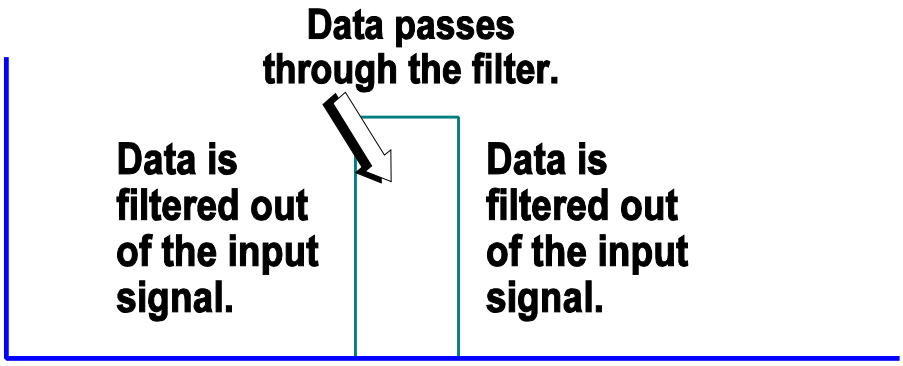
Low-Pass Filter



As shown above, the low-pass filter removes data above the selected frequency. Use this filter during low-speed (≤600 RPM) frequency analysis. Data collection times can exceed 2 to 10 minutes per average. Therefore, you can increase data collection times with a low-pass filter by retaining only those events that take place in the selected frequency range.

Band-Pass Filter



There are two types of bandwidth filters – Constant Percentage Bandwidth and Constant Bandwidth. These filters primarily serve the same function. The Constant Percentage Bandwidth filter changes width depending on the selected frequency.

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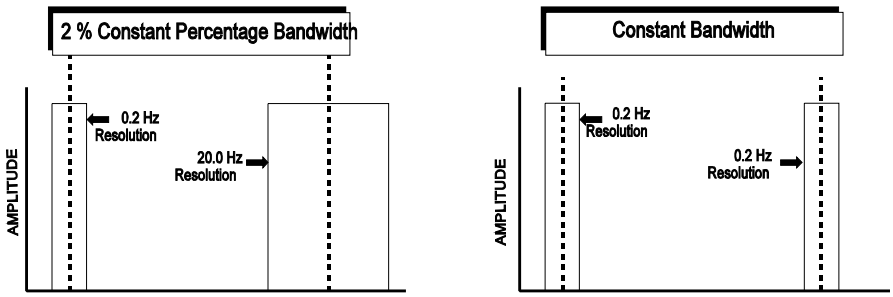
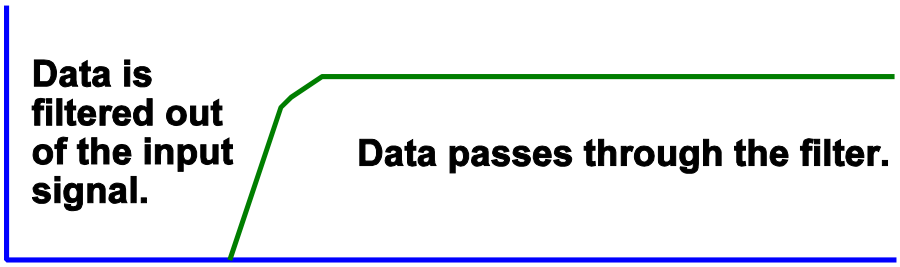
Idealized Bandwidth Filter Types



Notice the difference between the two types of filters in the example above. As shown, the filter to choose is the Constant Bandwidth Filter, because it provides the best resolution between both high- and low-frequency components.

High-Pass Filter



The high-pass filter gives you the ability to filter out low-frequency components for detailed analysis as illustrated above. This proves useful when low-frequency, high-amplitude data swamps the high-frequency, low-amplitude data you want to see. This situation often occurs when high-frequency events appear in the same plot as Run Speed and its relative harmonics.

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Frequency Demodulation

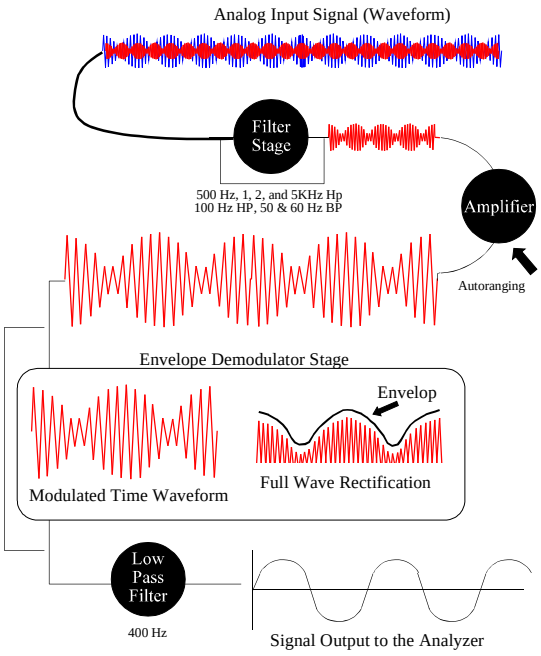
Several publications have described the application of demodulation to vibration measurements for machinery defect analysis. As an application tool, demodulation proves helpful in a wide variety of applications.

Demodulation primarily increases the effective dynamic range of the analyzer for certain types of low-level measurements. This increased range enhances defect indicators for fault analysis.

The 2120 **DEMODULATION** feature has the following advantages:

- Provides early detection of bearing and gear faults which might be masked by high amplitude, relatively lower frequency events.
- Lets you filter input signals for specific analysis requirements.
- Provides detection of faults which normal averaging techniques cannot detect.

Demodulation Process



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2120 DEMODULATE Lab

While collecting route data, it is common to have high amplitude frequencies dominate the spectrum. Often frequencies generated by unbalance, misalignment, vane pass and other relatively low frequencies “mask” the lower amplitude, high frequencies generated by rolling element bearing defects.

In gearboxes with meshing gear teeth, the Gear Mesh Frequency **(GMF)** has a tendency to mask the lower amplitude sidebands that are generated by various failure modes.

Even with the >90 dB dynamic range of the 2120, the bearing defects frequencies will may be lost in the noise floor.

The 2120 has a built in demodulator with a 1000 Hz default high pass filter.

When you choose **DEMODULATE**, the 2120 will automatically default to a **PREFILTER** setting of **1000 Hz.**

If you choose the **FREQUENCY** (F_{max}) greater than 1000 Hz the 2120 will automatically set the **PREFILTER** to the next highest filter setting above the **FREQUENCY** setting.

The 2120 has preset high pass filter (**FREQUENCY**) settings of:

20, 50, 100, 200, 400, 500, 1000, 2000, and 5000 Hz.

Another tool to help in the analysis of bearings and gears is to use the **Demodulation** pre-processing.

With the **ANALYZE** function, we will introduce the operation of this function in this chapter, and give specific setup considerations in later chapters as appropriate.

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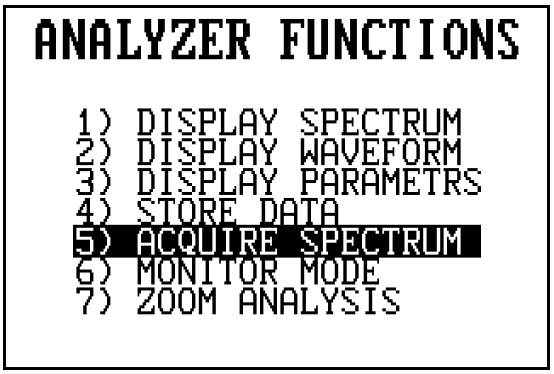
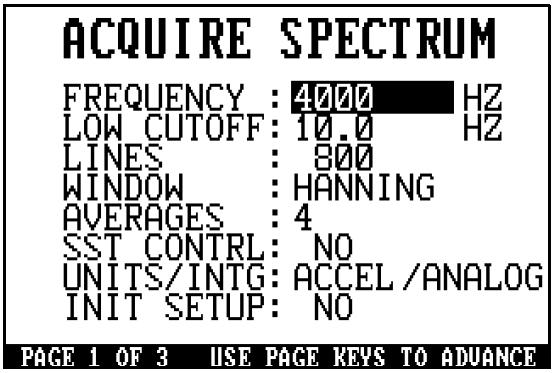
First we will look at a simple application of demodulation. We will start by collecting a normal spectrum from a variable speed rotor kit turning at approximately 3600 RPM (60 Hz).





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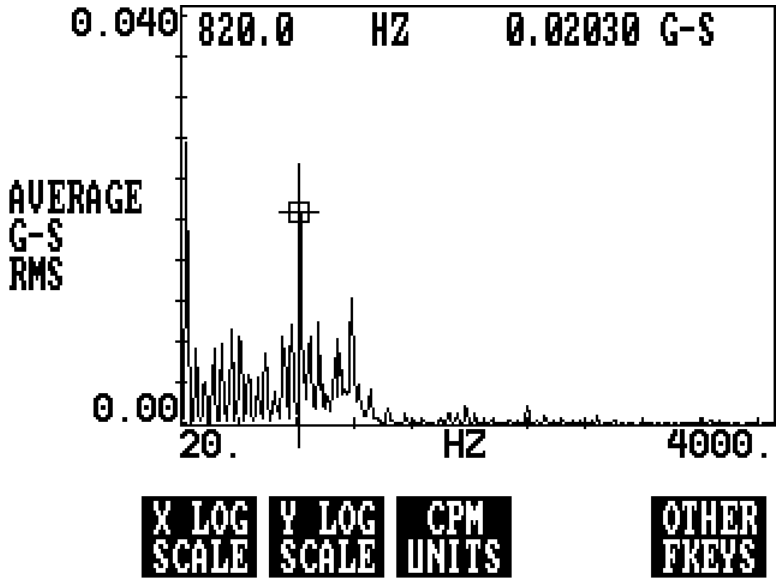

Normal Spectrum-No Demodulation

Displayed above is a Normal Spectrum - with no Demodulation.

The results are shown above. Notice in this example we have a peak at 820 Hz surrounded by many lower amplitude peaks. This is most likely an electrical forcing function related to the drive/motor.

This peak has the appearance of being modulated by another frequency because of the possible sidebands around it. We will now use demodulation to get rid of the high amplitude lower frequencies, as well as the carrier frequency (820 Hz in this example) and show the lower frequency modulator (with frequency equal to spacing of sidebands).

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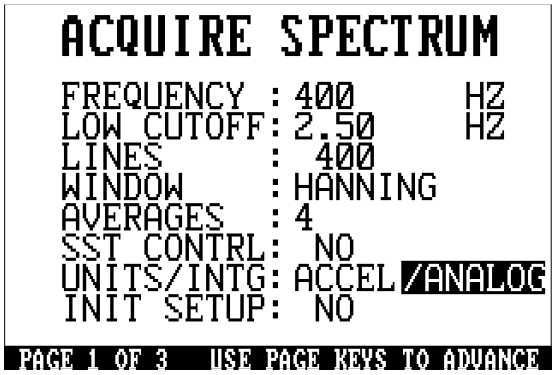
SETUP FOR DEMODULATION:

On **PAGE 1 OF 3** the **FREQUENCY** chosen will be the F_{max} of the spectrum you want to view. We will use 400 Hz. The **LOW CUTOFF** and **LINES** will be chosen as appropriate. Set: **HANNING WINDOW, 4 AVERAGES**, and **NO SST CONTROL**

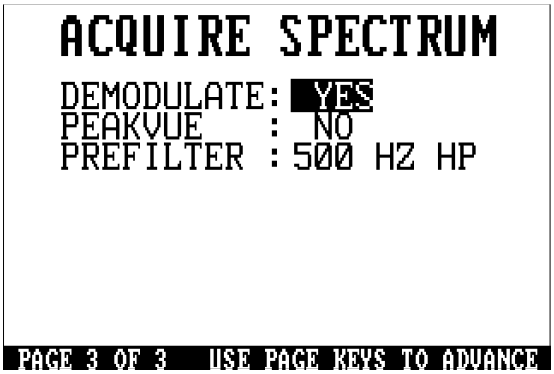


Set the UNITS/INTG: ACCEL/ANALOG. This puts the spectrum in g's acceleration which is important for proper demodulation operation. The analog integration mode will extend the dynamic range to the maximum. Press the **PAGE UP** key.

Page 2 will not be used.

Press the **PAGE UP** key to advance to **PAGE 3 OF 3**.

Toggle **DEMODULATE**: **YES**. You will see the **PREFILTER: 500 HZ HP** automatically chosen. This filter will default to a value based on your **FREQUENCY** setting on the 400 Hz F_{max} setting on page 1.
Press enter to collect data.



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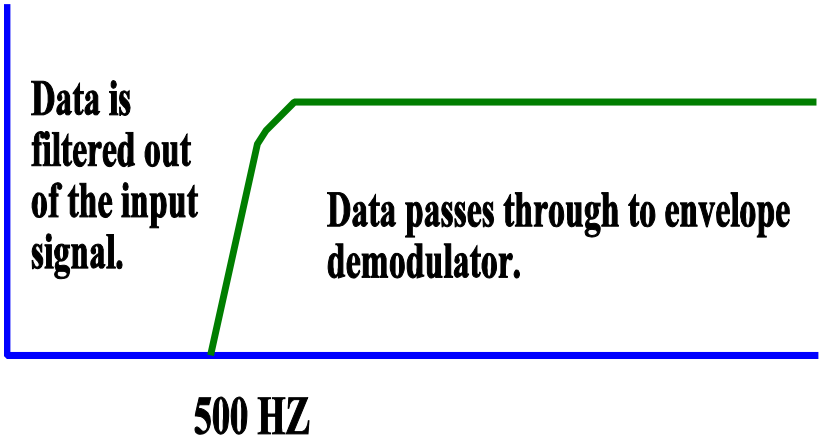
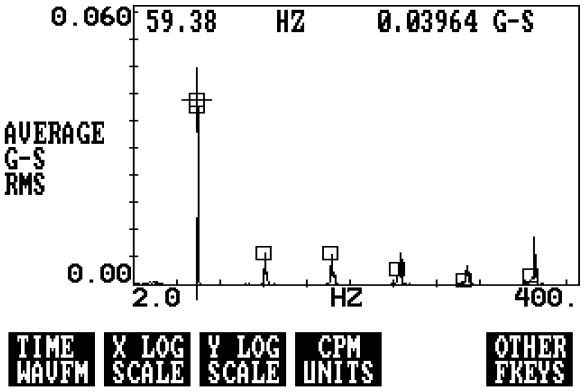
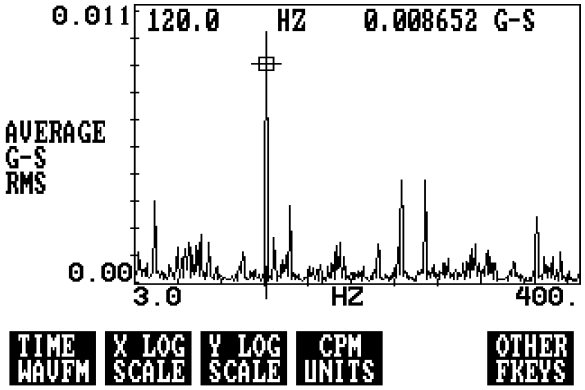




Normal Spectrum 0-400 Hz



Demodulated Spectrum

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Note: Both the Normal Spectrum and Demodulated Spectrum was collected with a 400 Hz F_{max}.

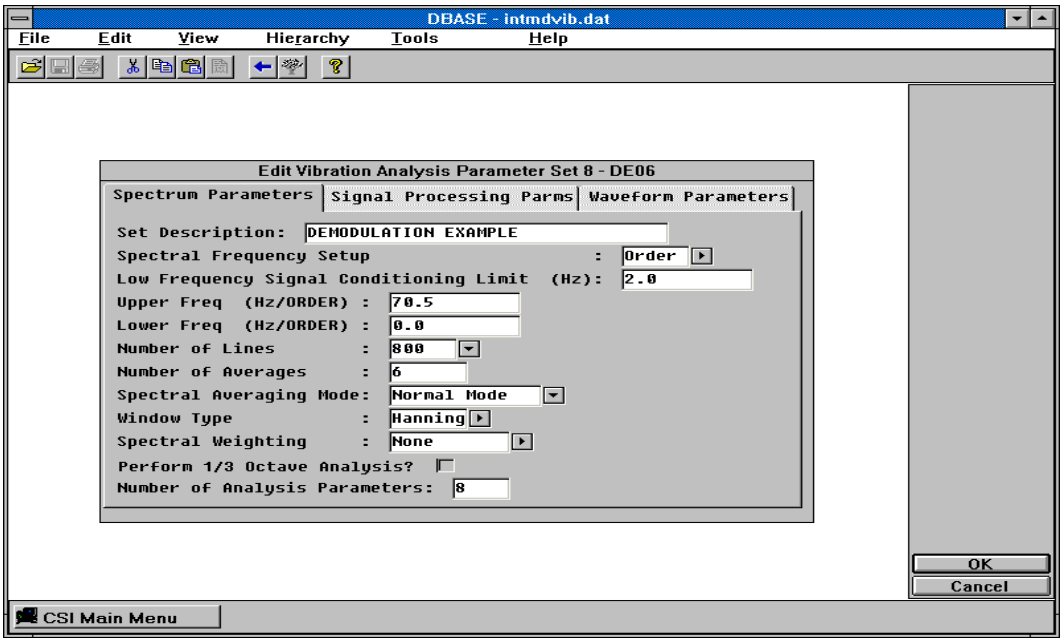
The normal spectrum (top) has a 1xTS peak (59.38 hz) with low levels of harmonics and not much else.

The demodulated spectrum has filtered off the high amplitude, low frequencies (1x and harmonics) and now shows us the possible modulator of 120 Hz. The 120 Hz did not show up in the original spectrum because the amplitude was much lower than 1x, 2x etc.

We will show how this feature is very valuable in bearing defect detection in the Rolling Element Bearing section.

MasterTrend Setup - Demodulation

Demodulation is a very valuable processing technique for route trending. Demodulation can help in difficult vibration environments such as slow speeds or where there is a large interfering signal (1x, 2x, GMF etc.). Here is a basic MasterTrend setup. The HP filter choices and F_{max} settings depend on the particular vibration environment.



Note: The Signal **Processing Parms** tab is where Demodulation is selected.

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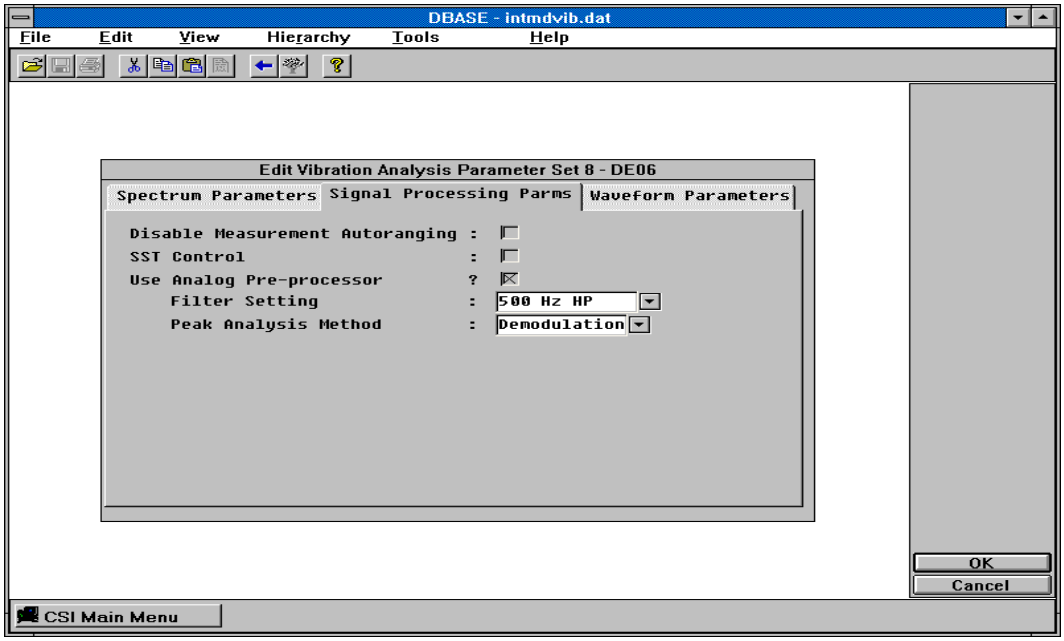
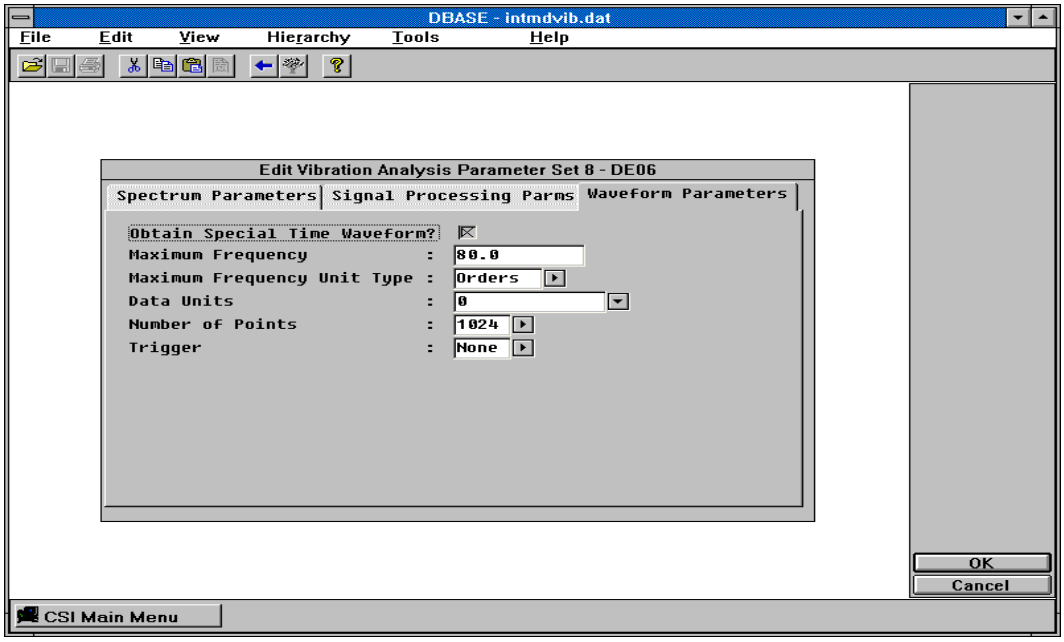


You will also need to collect a special time waveform because the waveform is not usable with demodulation. Select the third tab, **Waveform Parameters**.



An order based set with 80 orders and 1024 points will store a time block which will include about 5 shaft rotations which is ideal for most rolling element bearing analysis.

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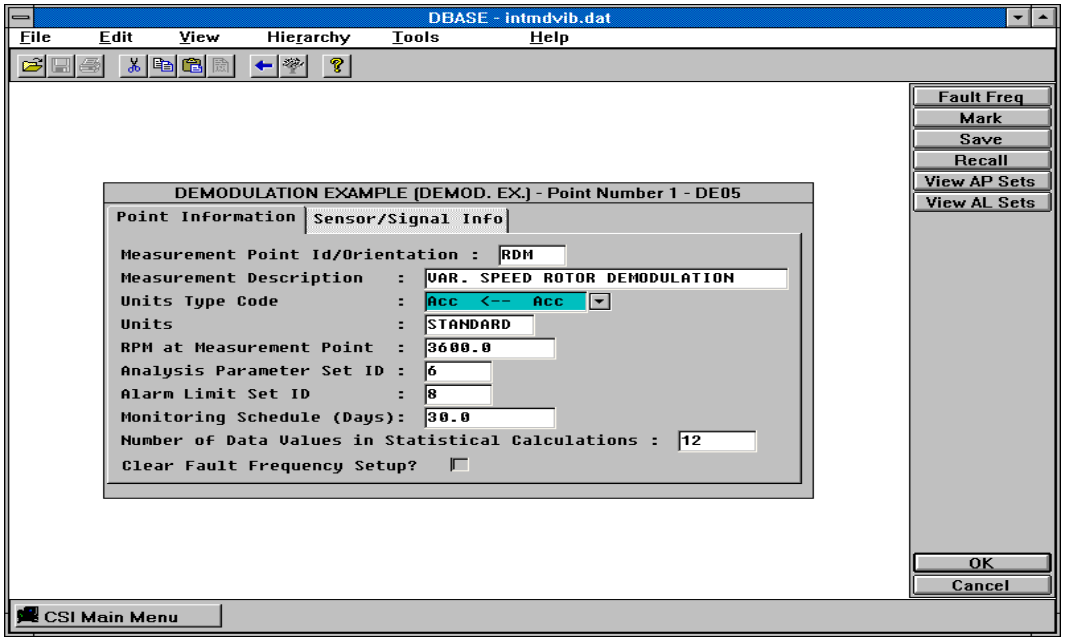

A measurement point setup is shown above. The **Units Type Code** should be setup for no integration (ACC:<---- ACC).

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Filter Stage

The Filter Stage contains eight selections:

- **five** high-pass filters, **two** band-pass filters, and a no-filter bypass option.

The choice of different filters lets you reject large signal components outside the frequency band you specify. Therefore, large components outside the specified frequency range do not overload the Model 2120 Machinery Analyzer, which lets it allocate more of its dynamic range to the band of interest.

The proper selection of a filter requires some knowledge of the frequencies you expect for the desired signals. For example, you know the desired carrier signal falls in the frequency range between 2,500 and 4,000 Hz (e.g., an early bearing signature). You also have a large, unwanted signal between 400 and 500 Hz (e.g., a **gearmesh**). For an appropriate filter setting, choose either the 1,000 or 2,000 Hz high-pass filter.

Now change the signal frequency ranges to between 2,000 and 4,000 Hz for the desired range and between 400 and 900 Hz as the unwanted range. The filter setting choice now becomes less clear. The 1,000-Hz filter provides only a small reduction of the 900-Hz signal. The 2,000-Hz filter reduces the desired 2,000-Hz signal somewhat. In this case, **try both filters** and select the one that gives you the best data.

If you undertake gearmesh demodulation, choose a **high-pass filter** that passes the gearmesh frequency and attenuates frequencies below gearmesh. If you do not know in advance the frequencies of the signals, select the bypass mode in the Model 2115/2120 Machinery Analyzer. Examine all available signals to determine filter selection.

Note: You will need to play with some of the filter-settings on your own and examine the data to see what filters will obtain the data you need to analyze a specific problem.

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Band Pass Filters

Use band pass filters primarily to examine electrical line frequencies. These filters reject all signals that are not within a band around the line frequency. You can also use band pass filters to detect the running speed of an induction motor (four poles or more). Simply set the instrument to demodulate the line current.

Demodulator Stage

The demodulator stage accepts as a portion of its input a signal known as the **carrier**.

The output of the demodulator is a signal known as the baseband signal (data relative to the envelope). The baseband signal contains the information you want.

Typical **carriers** in machinery applications include:

- gearmesh frequencies
- structural resonances
- bearing resonances
- electrical line current
- in some cases, the resonance of the sensor itself

The last case (sensor resonance) most often occurs when using the handheld probe. You must also have a mechanism in the machine that causes the information you want to be impressed upon or modulate the carrier. Fortunately, this latter condition is very common in machinery.

Low Pass Filter Stage

The final portion of the demodulation process is a **low-pass filter**.

The F_{max} setting effectively chooses this filter. This filter removes the residual carrier component that otherwise dominates the baseband output. This filter works in conjunction with the **PREFILTER** setting chosen on page 3 of acquire spectrum.

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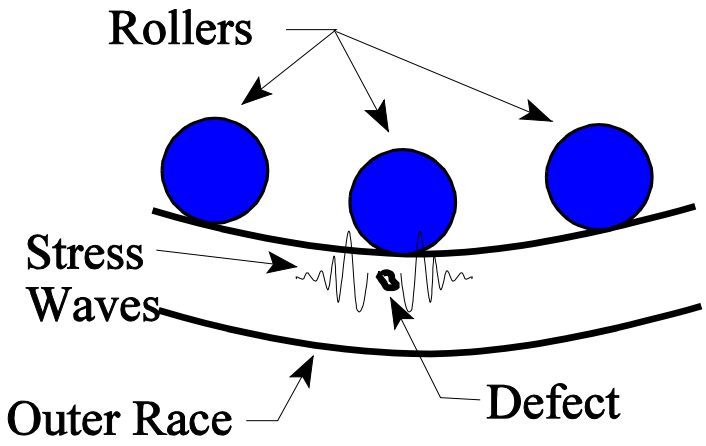
PeakVue

PeakVue stands for **peak value** and can be used to detect short duration higher frequency waves called **stress waves**.

Stress waves are created when metal is bent. For example, if a rolling element in a bearing passes over a defect which may still be below the surface of the inner or outer race, the race will deflect and then spring back.

This motion will create a sine wave. PeakVue measures the highest amplitude found in this waveform and holds it as the highest value during a waveform time length equal to 1 over the sampling rate.



PeakVue is a process that also passes the waveform data through a high pass filter to remove the rotational vibration frequencies from the data. However, filtering is its only real similarity to demodulation. The high pass filter should be set ***above*** the conventional $\mathbf{F_{max}}$ of the spectrum.

Generally, the 1000-Hz high pass filter is a good choice. The PeakVue process alters the length of short-duration, very high-frequency events occurring in a machine's vibration data. Be sure to set up a measurement point for this analysis method that does not integrate the data.

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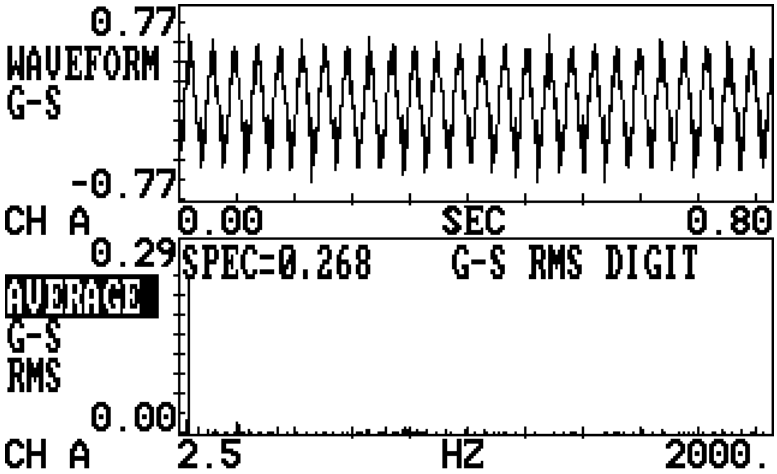
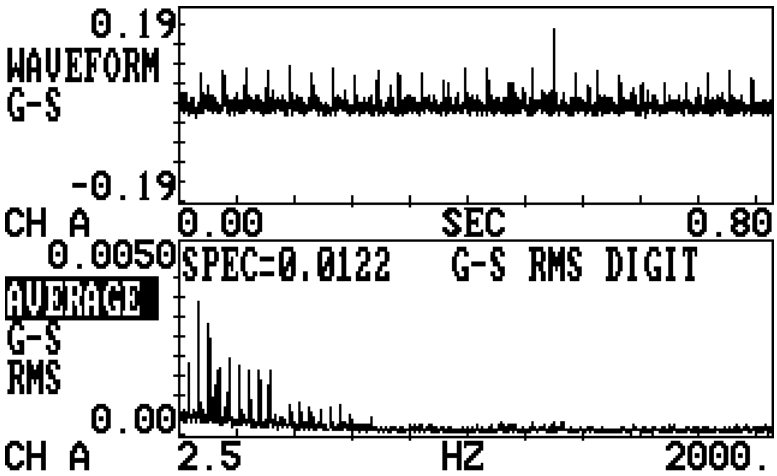


Data taken without PeakVue enabled.



Data taken with PeakVue enabled.

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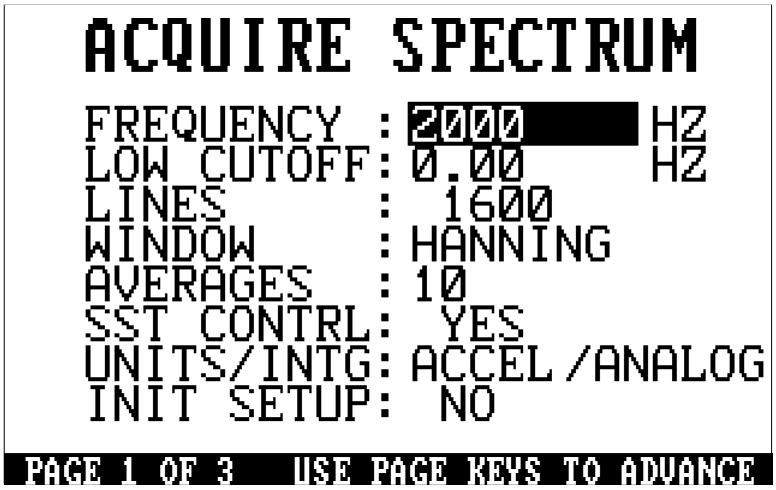
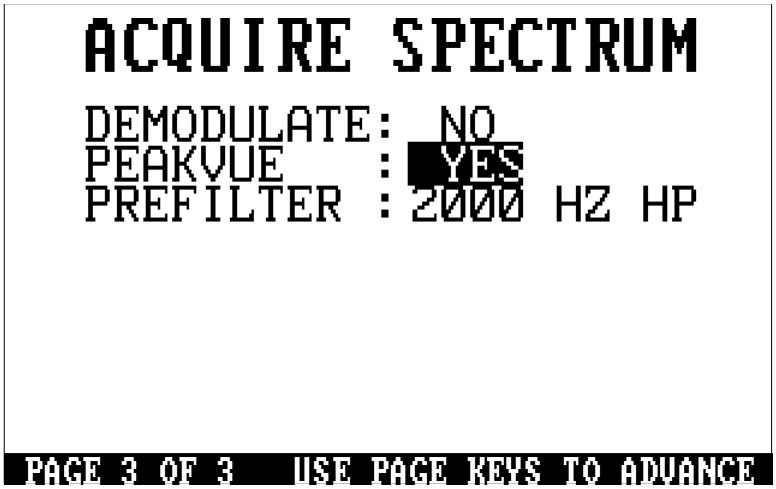
2120 Setup

As with demodulation, PeakVue can be accessed from the 2120 Acquire Spectrum Menu.



Toggle to the third page of the setup menu.



Collection of at least one data point per machine component is recommended using the **PeakVue** technology for early detection of bearing and gear defects as well as other machine problems.

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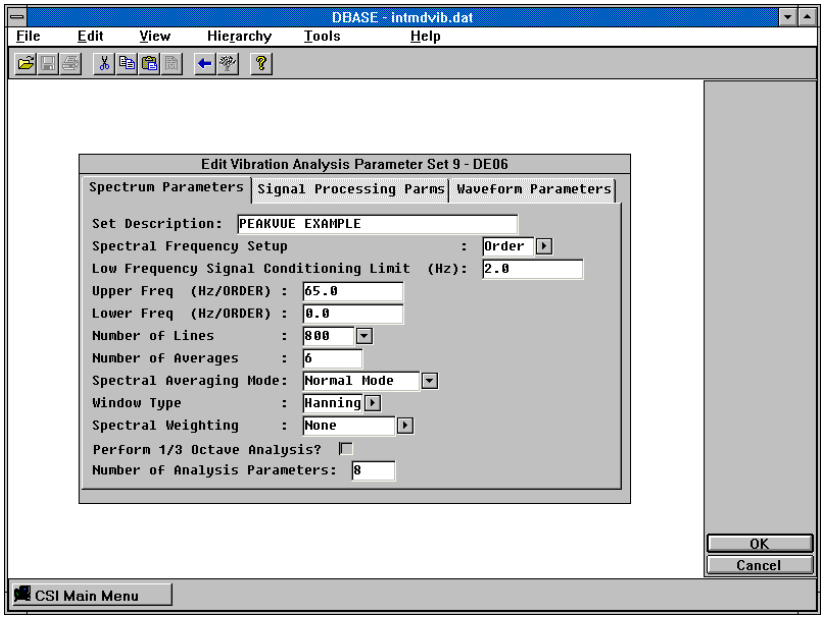
MasterTrend Setup

This can also be activated from a MasterTrend Analysis Parameter Set definition. In this case the F_{max} is set to 65 orders of turning speed.



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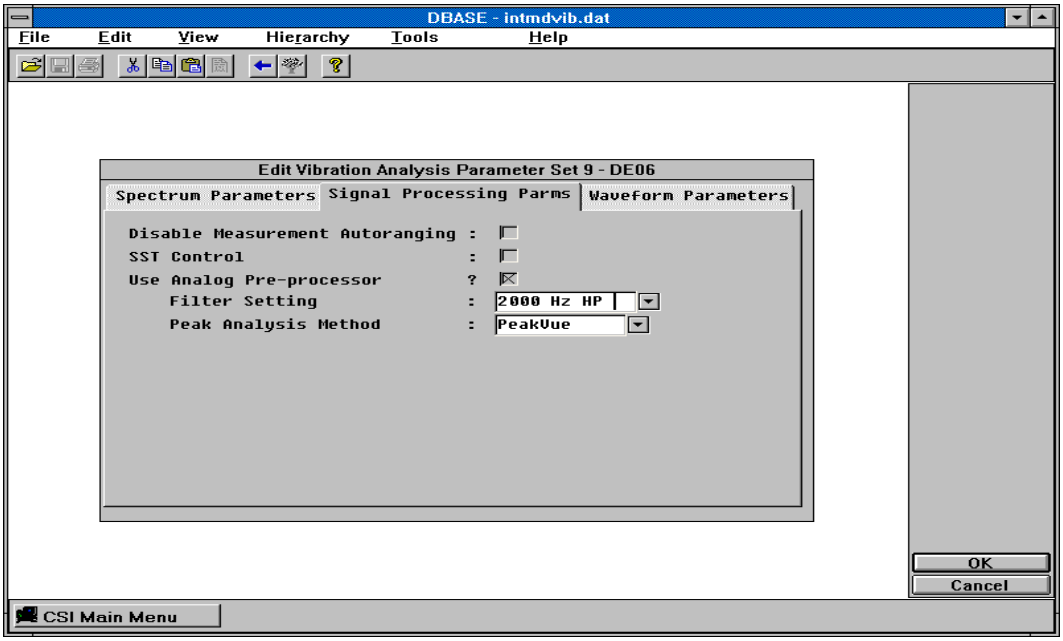
If this parameter set is used on a machine turning 1800 RPM.

The F_{max} will be 1800 CPM * 65 orders = 117,000 CPM.

If we divide by 60 to get the F_{max} in hertz, the F_{max} is 1950 Hz.

The **PeakVue Filter** should be set **above the F_{max},** so we will choose a PeakVue filter of **2000 Hz.**



Summary

These two techniques are very powerful early detection methods.

In this section, we have examined the differences between **Demodulation** and **PeakVue** as well as methods of enabling these features from both the 2120 Machinery Analyzer and MasterTrend (use of defined AP-Sets).

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Demodulation and PeakVue Lab

❶ Create a database: INTCLAS.DAT.

❷ Create a station: Signal Processing.

❸ Create a machine: Condensate Pump #1.

❹ Build five motor and five pump measurement points.

❺ Create Analysis Parameter Set Number 20. Enable the demodulation of data above 1000 Hz and make sure this set is assigned to the Motor Vertical points of the Condensate Pump #1.

❻ Create Analysis Parameter Set Number 21. Enable the PeakVue feature to process data above 1000 Hz and make sure this set is assigned to the last pump point on the Condensate Pump #1.

❼ Create a route: Pump #1.

❽ Load the route and collect your vibration data.

❾ **Do not dump the data back into MasterTrend yet.**

Summary

The key to collecting worthwhile data is understanding how the analyzer performs its job.

When setting up for data collection, remember that the:

- spectral F_{max}
- the lines of resolution
- the waveform T_{max}
- and the waveform size are all related

Don't forget to choose the correct analysis window for each measurement application. The conversion from analog data to digital data can alter the way the data appears, so remember to recognize the limitations of the measurement process. Using the information in this section should help unlock the mysteries of digital signal processing.

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Section Ten
Waveform Characteristics

Objectives

- Describe five waveform characteristics.
- Identify waveform symmetry using an APD.
- Discuss waveform modulation and how it translates to the frequency domain.

General Description

Waveform data has been an area of neglect since many analysts have not had the fundamental training to properly use the waveform for diagnostic purposes.

This section discusses some of the fundamentals of **time domain** data. The time domain (waveform) data can be very valuable to the analyst, especially with complex machines.

A number of different displays use the time domain. Displays such as **synchronous time averaged** data is averaged in the time domain. APD (Amplitude Problability Distribution) this is a function of Wavepak, displays the symmetry and skewness of the waveform signal.

Orbits use two waveforms from transducers or eddy current probes mounted 90° apart and show the shaft rotational characteristics. These displays and others will be discussed in detail later in this section.

Each defect type has a characteristic waveform, which subsequently translates to the frequency domain.

However, there are characteristics and specific events that **do not** translate to the frequency domain as discrete peaks. In order to truly understand this limitation, the analyst must first understand how the time domain data is gathered and transformed into a spectrum through the Fast Fourier Transform (FFT) process.

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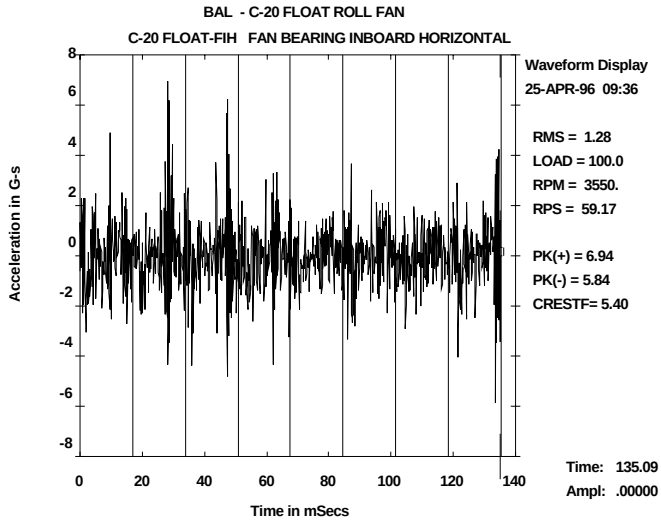
What Is A Waveform?

Time domain data, raw transducer output, signal voltage and many other terms refer to waveforms. The difference relates to the input transducer type, the axes being considered (magnitude vs. time), and whether or not the signal has been manipulated with a digital signal analyzer. **Historically, waveform data has been gathered using analog systems.**

Waveform or time domain data is comprised of **amplitude with respect to time**. Signals with an amplitude, whether vibration, current, voltage changes, or other signal types, change with time. The time change may be slow (1 to 10 minutes) or extremely fast (1 to 200 msec). In either case, time is a primary aspect of understanding waveform.

Waveform Characteristics

There are certain things to look for when conducting waveform analysis. In order to know what to look for, we should discuss the characteristics and what they mean to the analyst.



As was previously stated in this section, the waveform provides specific characteristics for defects of a single or multiple nature.

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Those machines containing multiple defects are much more complex and require additional time during the analysis procedure.

Note: The waveform is only as good as its definition. If the resolution of your waveform lacks definition, the data can be worthless, or poor at best.

Once the characteristics have been properly identified, the analyst can rule out certain fault types.

For example:

If a waveform is periodic (sinusoidal)

- looseness
- cracks
- resonance
- antifriction bearings

Could probably be ruled out. You may not know what the problem is, but you know what it is not.

Listed below are waveform characteristics an analyst should look for when analyzing the waveform:

- Amplitude
- Asymmetry
- Electrical v Mechanical
- Distortions
- Periodic
- Spikes/Impacts
- Non-Periodic
- Modulation
- Complexity
- Discontinuities
- Low Frequency Events
- Truncation/Restrictions to Motion

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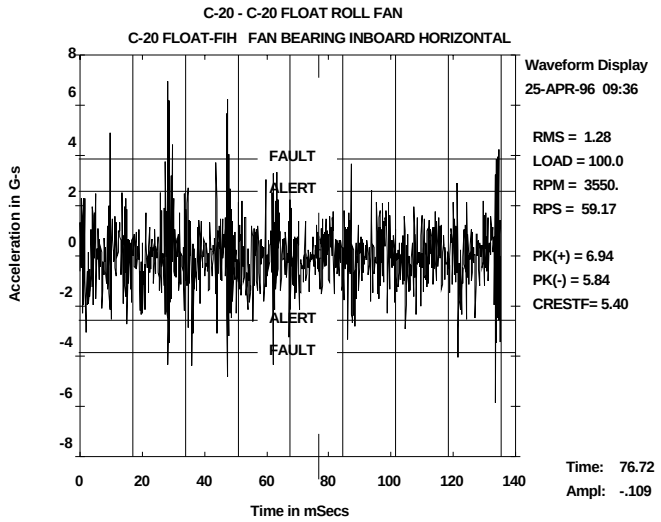

Amplitude

When diagnosing machinery faults using the time waveform, similar to spectral data, we are concerned with the amplitude of the waveform.

When we are discussing bearing and gear waveforms, we use the peak to peak amplitude of the waveform. This is often referred to as g swing.

The g swing is the sum of the absolute value of the maximum positive and negative amplitude in that period.

MasterTrend calculates this value and gives us the ability to trend and alarm based on this and other waveform values.

PK(+) = 6.94

PK(-) = 5.84

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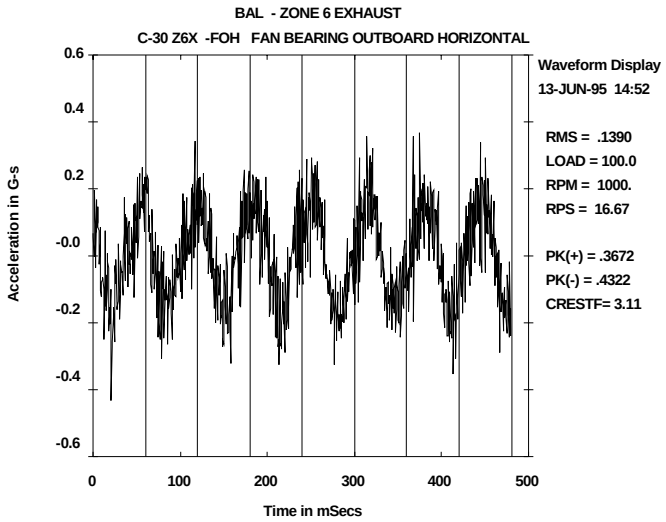

VA2CHSTY.DAT, BAL-IMBALANCE, C-30 Z6X, ZONE 6 EXHAUST

Periodic

Sometimes referred to as a **deterministic simple signal**, this is an ideal signal which repeats itself exactly after a fixed period.

This is not possible in the real world. However, there are some machinery faults which have this characteristic.

A single plane balance problem will have a very periodic waveform due to the mass rotational center and the rotor shaft of other component center line differences.

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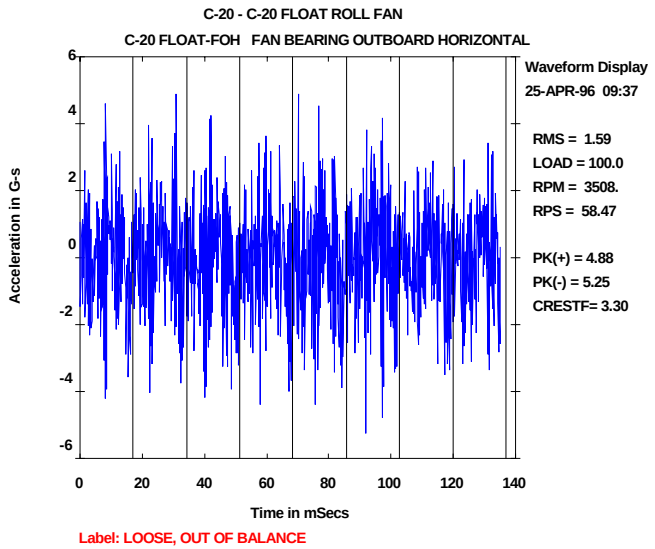

Complexity

To determine the complexity of the waveform, establish whether the signal is:

- periodic in nature
- estimate the harmonic content
- determine if the signal is synchronous
- non-synchronous
- identify whether the waveform correlates directly to the spectral data.

Confirmation will be covered in detail later in this section.

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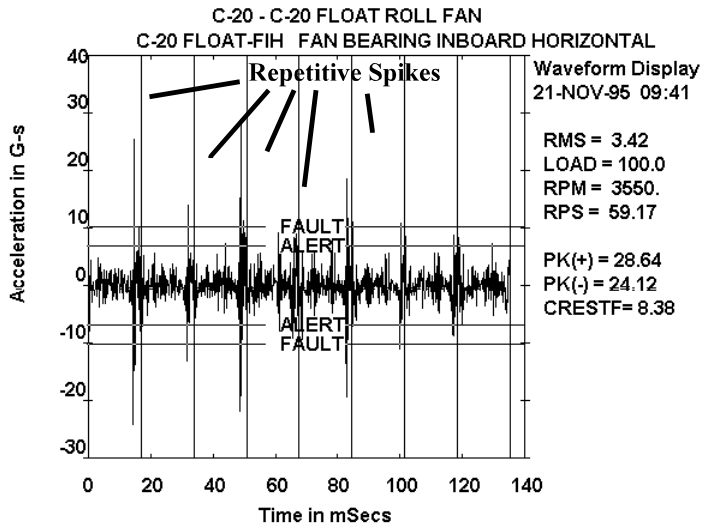

Impacts/Spikes

Impacts or Spikes may or may not be repetitive in nature.

The non repetitive spikes generate white noise. In other words, they excite all of the frequencies present.

Repetitive impacts or spikes, such as those produced by rolling element bearing defects or broken gear teeth, may excite discrete frequencies and therefore show up well in the spectrum. The excitation is proportional to the energy induced by the impact.

This characteristic is best detected by defining a waveform amplitude type in **acceleration.** Acceleration data is **proportional to force**, and the sudden impact induces a force which could be of grave importance.

The **crest factor**, which is equal to the maximum peak **(positive or negative) divided by the RMS of the waveform**, is a good indicator of the impacting. This value can be setup as an analysis parameter and trended in MasterTrend.

PK(+) = 28.64
PK(-) = 24.12
CRESTF= 8.38

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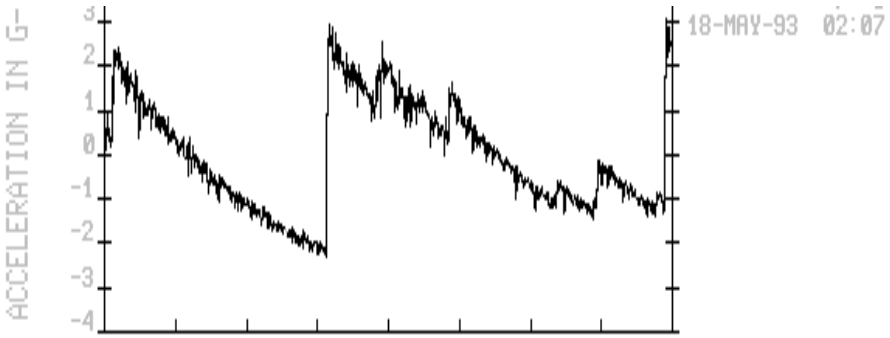

Discontinuities

This characteristic is usually associated with faulty equipment due to the discontinuous nature of the data.

Data with this characteristic has breaks in the data where there appears to be a loss of input signal or a significant increase/decrease in amplitude.

This is not a uniform change such as resonance, load changes, or even sudden component failures.

Discontinuous data is typically unpredictable, and very distinct.

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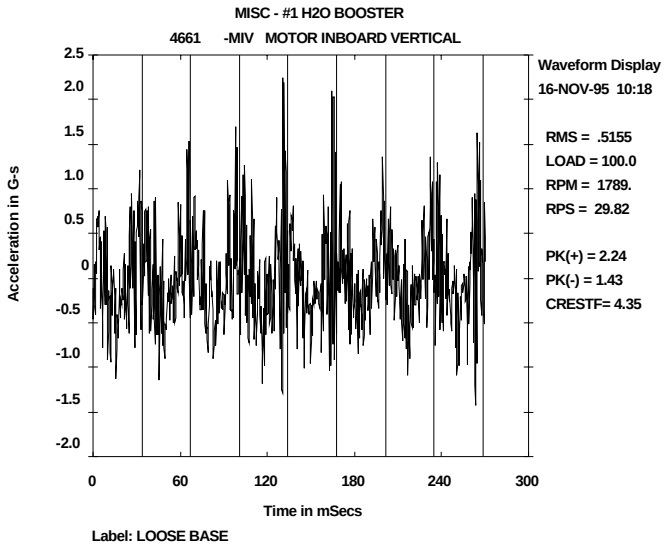

VA2CHSTY.DAT, MISC, #1 H2O BOOSTER

Asymmetry

Asymmetry refers to the relationship between the positive and negative energy.

A waveform is **asymmetric** when there is more energy in the positive plane than the negative or vice versa.

Notice above, the waveform display shows the peak positive/negative energy. We can use this information to make some assumptions about our data.

Asymmetry refers to the direction of movement relative to the transducer mounting with a ***positive*** signal representing energy ***into*** (towards) the accelerometer and a ***negative*** signal representing ***away***.

A tool which is designed to check this type of characteristic is the APD, Amplitude Probability Distribution.

PK(+) = 2.24
PK(-) = 1.43

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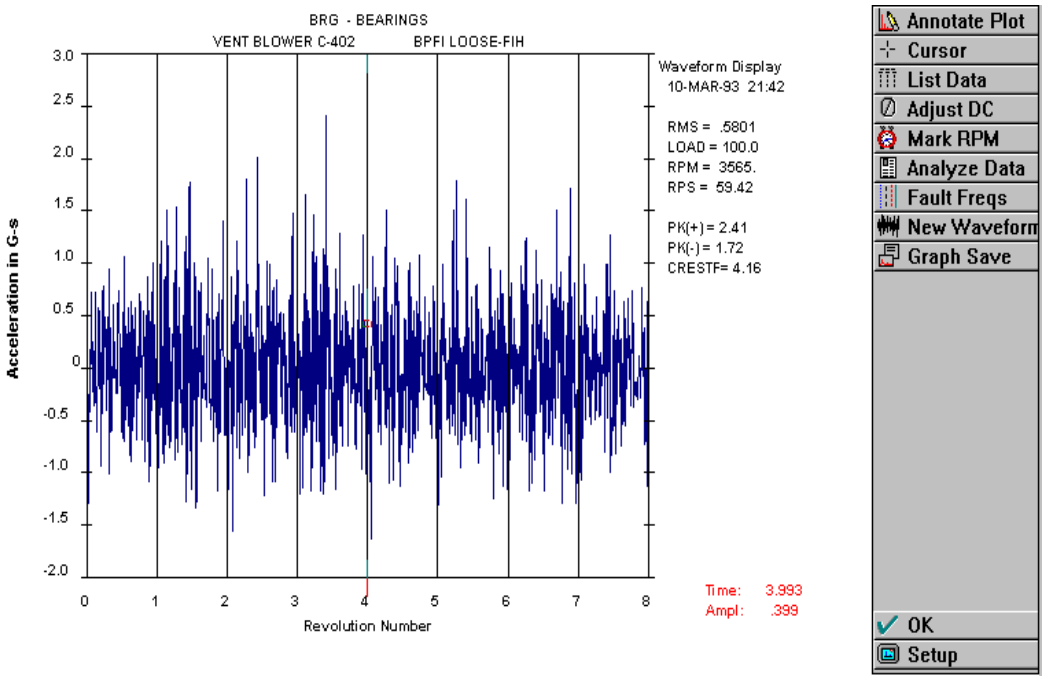
Also a tool that can be utilized is to select the Analyze Data feature in Diagnostics Plotting when in Waveform Analysis. Select option from Graphics Function Bar.




```
                    ANALYSIS OF WAVEFORM
                    ********************
Machine:       (BRG ) VENT BLOWER C-402
Meas. Point:   BPFI LOOSE-FIH  -->  PILLOW BLOCK INBOARD HORIZONTAL
Date/Time:     10-MAR-93  21:42:48    RPM=  3565.   Units=G-s

 WAVEFORM % DISTRIBUTION (15 Intervals) - Acceleration in G-s
                          *   *
                          *   *
                          *   *   *
                      *   *   *   *
                      *   *   *   *   *
                      *   *   *   *   *
                  *   *   *   *   *   *   *
              *   *   *   *   *   *   *   *   *
-------------------------------+-------------------------------
   0   0   0   2   5  14  21  21  17  12   4   1   1   0   0  (%)

(-)   Max Peak   (+)   (-) Crest Factor (+)   (Dif) Symmetry (Abs)
--------------------   --------------------   --------------------
 -1.72         2.41     2.97          4.16     -6.1           8.4

(-)   Ave Peak   (+)   (-) RMS Amplitude (+)   Kurtosis   Skewness
--------------------   ---------------------   --------   --------
 -.8379       .9173     .5532         .6063      .157       .145
```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Amplitude Probability Distribution

An APD or Amplitude Probability Distribution is similar to a Hystorgram.

The signal is broken down into amplitude percentages, and then the amplitude is plotted.

The X-Axis is the amplitude and the Y-Axis is the percentage of the signal that falls into that amplitude range.

This can be extremely helpful in finding the root cause of a problem.

The APD is typically used for acoustical analysis.

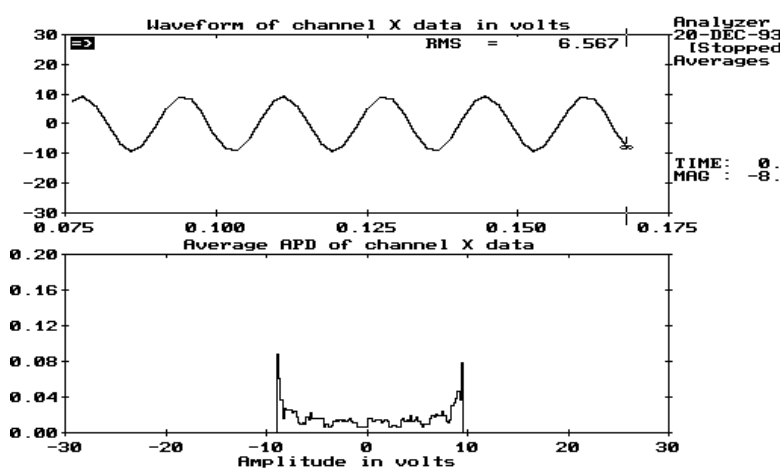
It can also be used for machine vibration analysis to find the balance of the signal (asymmetries), the direction, and possibly the location of a specific defect especially those that may not stand out in the waveform or the spectrum.

Sinewaves

Sinewaves are very symmetrical, which means there is a balance of energy in the positive and negative planes.

If most of the vibration signal is evenly distributed and sinusoidal, there is a strong possibility it is due to a synchronous component such as imbalance, misalignment, gears, blades, etc.

The waveform and APD below show the shape of a sinewave and the probability related to this type of signal.



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Notice that the APD at the bottom of the above display shows a set of peaks at the maximum and minimum amplitude locations. Each of the peaks represent the total probability of the signal in that location.

This could also be called a Hysteresis display. The probability of the signal being in the ± 10 volt location is much more probable that the signal being at the zero location of the display.

This information lets the analyst check for the direction of motion and the skewness of the data. Looseness and other machinery faults may be identified easier if the APD is used with the waveform and spectrum displays.

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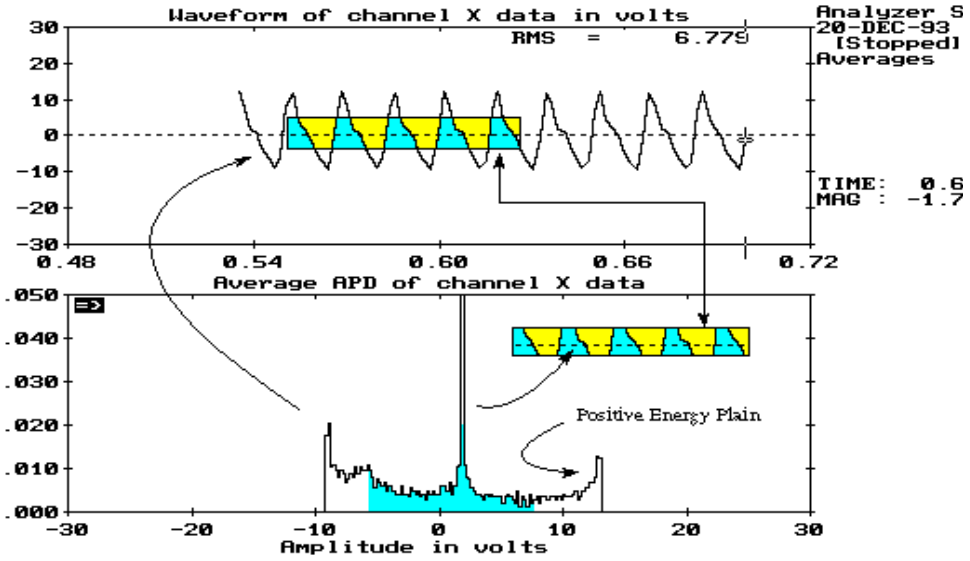
Triangle Wave

With a triangle wave, we see the relationship of the waveform and a different type of APD display.

The waveform takes an extended time to pass through just above the zero crossing of the waveform about the 1.5 - 2.0 volt location. Therefore, the APD shows a high probability that the signal is in the range of energy previously mentioned.

Also note that the data is skewed to the negative plane. Again, this provides the analyst with the direction of motion.



The previous illustration displays the direct relationship between the waveform and the APD. Bear in mind that the APD provides another tool to determine location, direction, and asymmetry.

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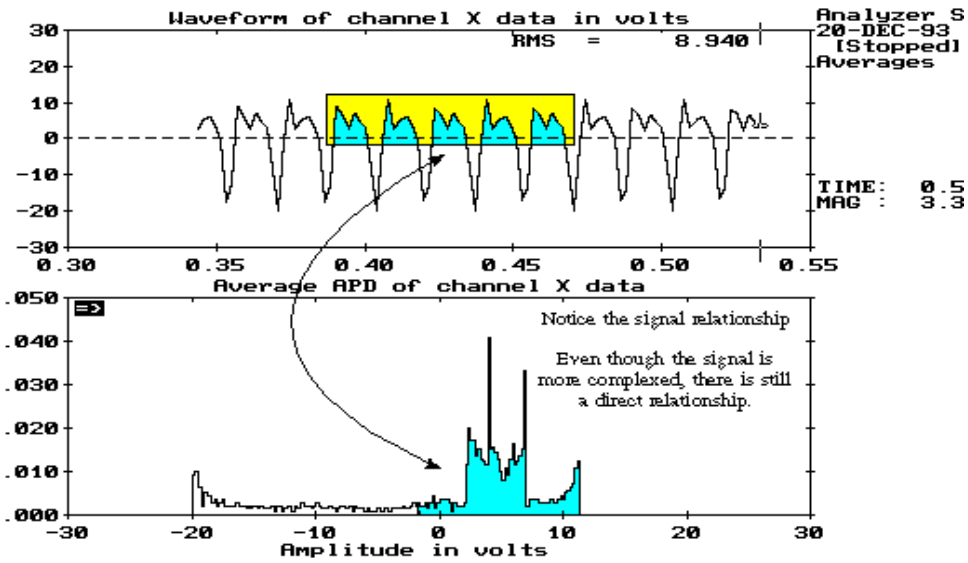
Squarewave

The squarewave below provides some insight into the use of the APD for checking asymmetries. Remember that ***symmetry*** refers to the balance of energy. Therefore, with a slightly more complex signal, this becomes more important especially when performing **Root Cause Failure Analysis**(RCFA).



In the illustration above, the signal is **asymmetric**, and there is more energy in the positive plane than the negative.

If this is a mounted transducer, such as an accelerometer, the analyst should look for more movement in the direction of the transducer.

The energy in the positive plane shows movement toward the transducer, and the negative plane is obviously the opposite.

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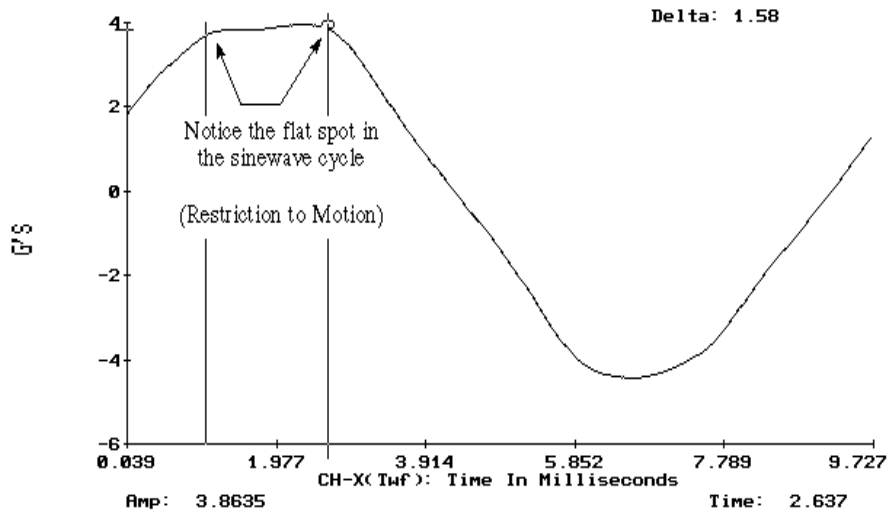
Truncation/Restrictions to Motion

Truncation means to abruptly shorten, or to appear to terminate.

In waveform analysis, this characteristic indicates restrictive motion. It is usually caused by a light rub. With enough waveform resolution and understanding of the phase characteristics of specific amplitude types, the analyst can locate and identify the severity of the rub. This is another application for multi-channel analysis, specifically orbits.



Modulation

All the waveform characteristics up to this point have dealt with signals of a constant amplitude.

A varying signal will cause the waveform to become modulated. The type of modulation occurring determines its classification. Commonly referred to as Beat frequencies, these may be broken into three specific categories.

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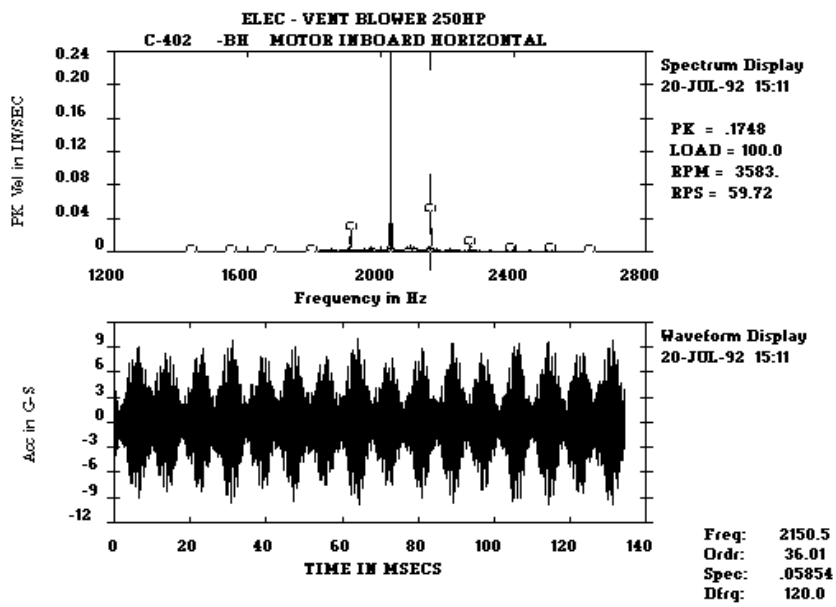
Amplitude

If the amplitude of a single frequency signal changes during each cycle, it is no longer periodic. If this signal is a single frequency and the amplitude varies periodically at a lower, but unrelated frequency, then the spectrum has an easily recognized pattern. The spectrum will have a peak at the signal's frequency with one peak on each side spaced at the frequency of the amplitude change. These peaks are referred to as ***sidebands*.**

Amplitude modulation is common when analyzing inner race bearing defects. This occurs when the defective bearing component passes in and out of the bearing load zone. The middle of the load zone is typically where the highest amplitudes in the waveform show up.



The spectrum and waveform show slot pass frequency from an AC induction motor. The primary signal at 34xTS is marked with a vertical line. The sideband cursors mark the amplitude change at 120 Hz.

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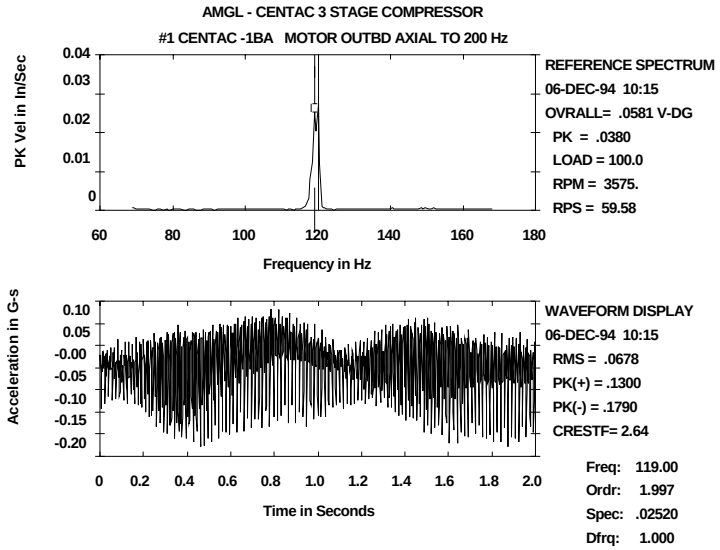
Beating

A beat pattern in the waveform is related to amplitude modulation. The difference is modulation caused by a beat is comprised of two unrelated single frequency signals, closely spaced in frequency. The signal's waveform appears like a modulated signal at the mean frequency of the two components, amplitude modulated at a frequency equal to their mean difference frequency.

Beating is often found in two pole induction AC motors. The close proximity of two times line frequency and the second harmonic of turning speed cause this beat. The two times line frequency component can be high in amplitude if the motor has a rotor or stator defect.



An example of beating is shown above. The 2x RPM and 2x line frequency are separated by less than .5 Hz. The waveform shows the amplitude modulation associated with beating.

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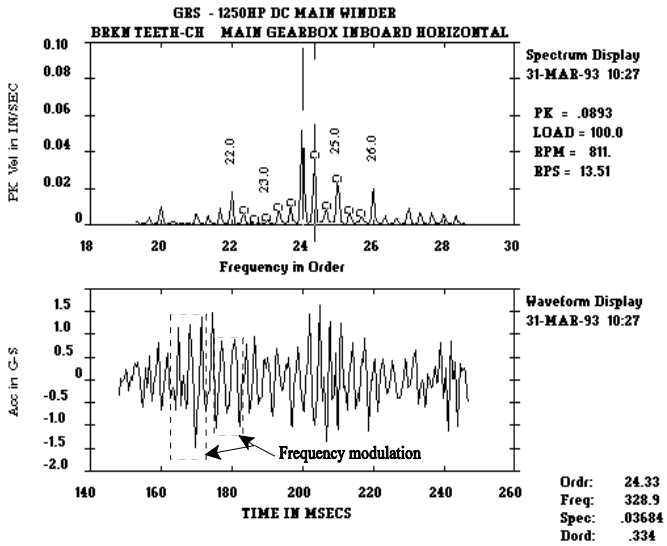
Frequency

Rarely seen in a routine environment, this is a change in frequency without a change in the signal amplitude. Frequency modulation typically occurs in gearmeshing vibration, due to the small speed fluctuations caused by tooth spacing errors and faults as they develop. Almost all effects which cause frequency modulation in gearboxes and other machines will also produce amplitude modulation. It may be difficult to separate out the two effects, but a very wide spread of sidebands in the spectrum is usually an indication that significant frequency modulation is present.



The example above is from a winder gearbox. The vertical line in the spectrum marks gearmesh frequency at 24xTS. The sideband cursors mark the output shaft speed with labels identifying sidebands spaced at input shaft speed. The waveform has been expanded to show the frequency modulation occurring. A good illustration is shown between 170 and 180 msecs. As stated above, amplitude modulation can also be seen.

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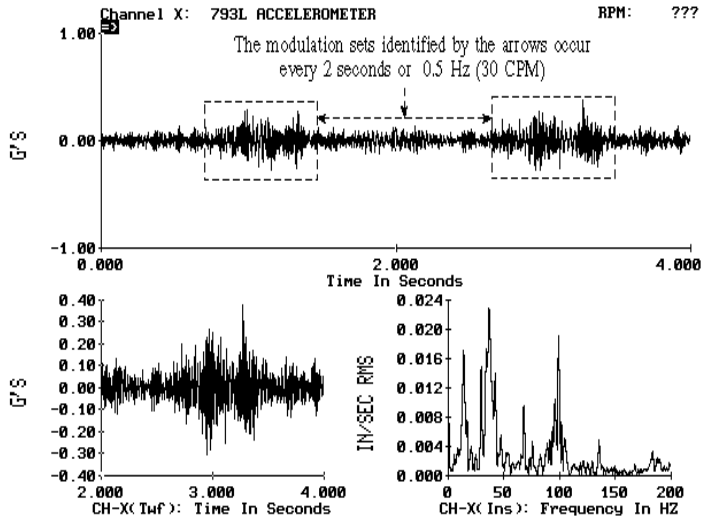
As seen from the examples showing modulation, the signal is usually made up of a combination of amplitude and frequency modulation. One thing is for sure, if the spectrum contains sidebands, the waveform will be modulated.

Now that modulation has been discussed, the remaining characteristics of the waveform can be presented. These characteristics build on the ability to identify modulation.



Low Frequency Events

When performing detailed analysis, you need to be able to collect and analyze data in excess of one minute for low frequency problems. This is extremely important when the machine in question has an operational speed below 200 RPM. The challenge in identifying low frequency defects is having sufficient time in the waveform. A low frequency event may only appear once in the collected time domain. As discussed earlier, this event will not be transformed into the spectrum.

The waveform data above displays an event that occurs approximately every two seconds or 0.5 Hz (30 CPM). Therefore, unless you were specifically looking for a low frequency event with a very low frequency setup using special transducers, you could have missed the event completely.

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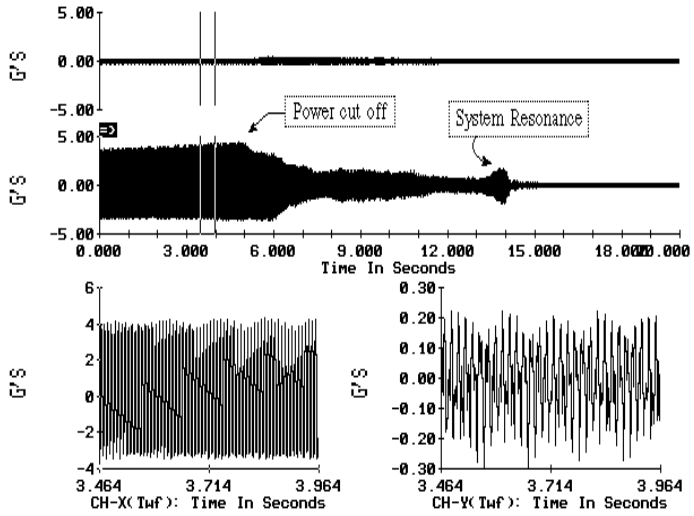

Electrical vs. Mechanical

Determining if the source of energy is mechanical or electrical is sometimes difficult. Appropriately set up waveforms can be a great help. Setting up for a long enough time to capture the operational conditions and the machine shutoff point can identify the source. The advantage of using the time domain as opposed to the frequency domain is there is no need to worry about the screen update time or sampling rate.

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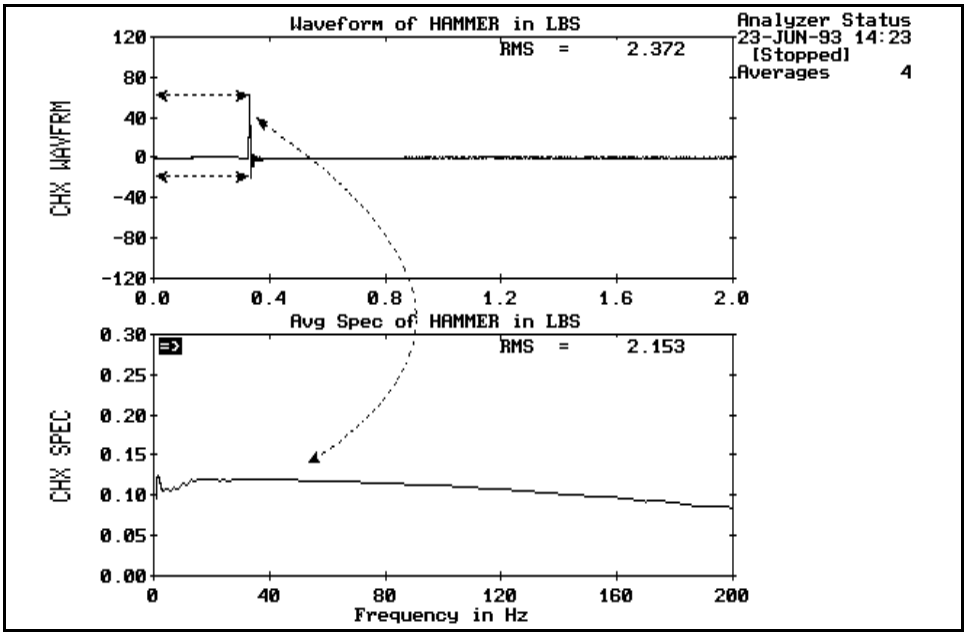
Waveform and Spectrum Relationship

Now that we have identified the specific characteristics of the waveform, let's compare the waveform and its associated spectrum.

For our purposes, each spectrum has an associated waveform. The spectrum is made of this waveform. As discussed earlier in this section, some of the characteristics in the waveform do not translate to the FFT due to the way the calculations are made. The assumption is that there is a repetitive cycle of events made up of sines and cosines. However, this is not actually the case.

If an event happens only once, then this event has no frequency; therefore, the spectral representation is a continuous spectrum. The data below shows the waveform and spectrum of an instrumented force hammer. Though used for multichannel impact testing, the characteristic differences are shown.



In the waveform above, there is no repetition in the event; therefore, there is no frequency. An impact with little or no time associated with it results in the spectrum above. The analyst input approximately 80 pounds of force, yet there is only 0.1 pounds of force at all frequencies ranging from 0 to 200 Hz with little deviation in the actual amplitude. This may cause the spectrum to look as if the problem has fixed itself. However, the data in the waveform is extremely severe. The waveform amplitude and spectral characteristics don't match because the defect may have become so nonsynchronous and nonrepetitive that there is no definable frequency content.

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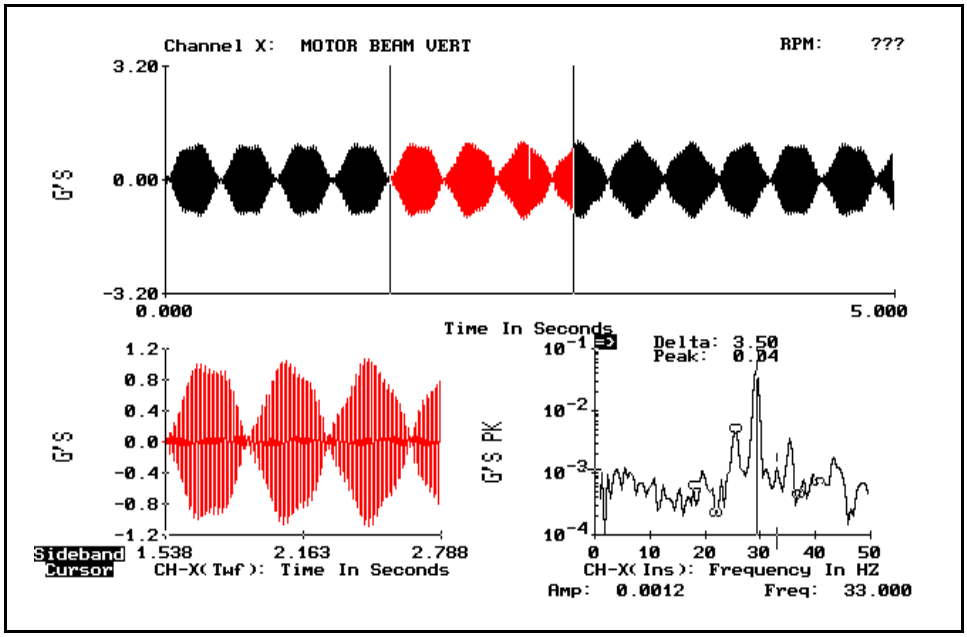
Modulated Waveforms

Finally, when modulation is involved, there is a direct relationship between the waveform and the spectrum depending on the differential time (Δt).

Knowledge of the modulation Δt helps determine the resolution required for detailed spectral analysis. Also, from our previous discussion on modulation, we know there is a carrier frequency that the modulation must follow.

Gears, bearings, and electrical defects each have carrier frequencies. For gears the carrier is the frequency where the gears mesh. However, a carrier frequency for an electrical defect could be the line frequency (F_L) or $2 * F_L$.



Believe it or not, the data above is real. Generated from a slow-speed application, this is a gear defect with sidebanding of the output gear speed.

There is a direct relationship between the waveform and spectrum, but in some cases it is not easy to identify.

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Waveform Analysis As Confirmation

One of the most functional uses of waveform analysis is to confirm the diagnosis of a particular fault type.

Every fault condition has a corresponding waveform characteristic.

Unbalance, for example, has a sinusoidal pattern with one major event per revolution.

Misalignment, which is primarily offset, typically has harmonic activity with the waveform having the same number of events per cycle as the spectral data has peaks. A misalignment condition generating a second and possibly a third order peak shows two or three sinewaves per revolution.

Looseness will have a complex waveform with many peaks within one revolution. This will confirm the spectral characteristics of multiple harmonics of turning speed.

The following examples show how to use waveform data to confirm existing spectral analysis.

We are not suggesting that the primary diagnosis be conducted in the time domain. But, it should be used to reassure the analyst of his/her judgement call.

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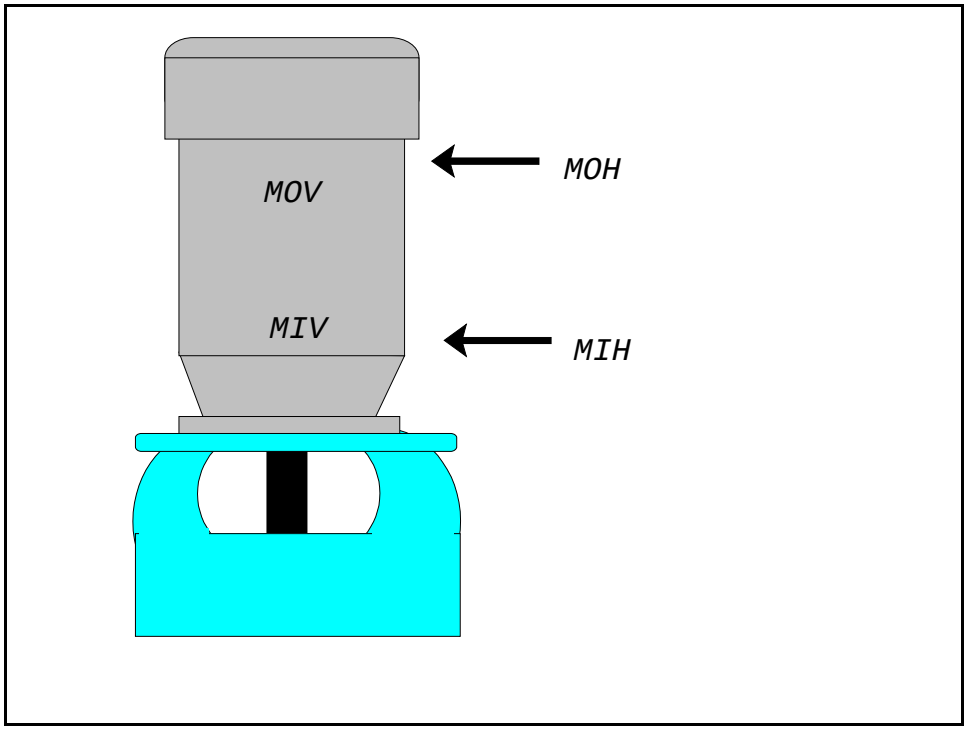
Vertical Turbine Pump Unbalance Example

The vertical pump motor bearing spectrums show unbalance as the major forcing function. The time waveform confirms the diagnosis. The motor was shaking so bad, the leads were chafing in the terminal box and grounding to the frame.



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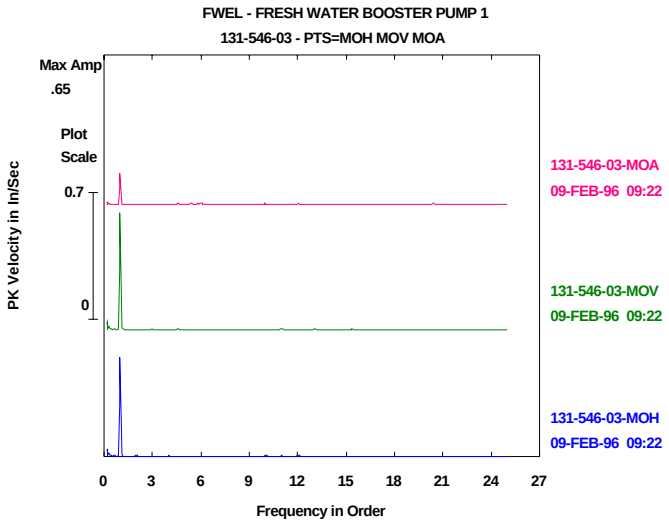
Multispectra Plot - Data Comparison

The multiple point spectrum plot below shows radial and axial measurements taken from the top of the vertical motor.



VA2CHSTY.DAT, BAL-IMBALANCE, 131-546-03, FRESH WATER BOOSTER PUMP 1

The spectrums show a dominant peak at 1X turning speed. The highest level is at the MOV position. Note the absence of harmonic activity and low level of non-synchronous energy. The radial directions are much higher than the axial.

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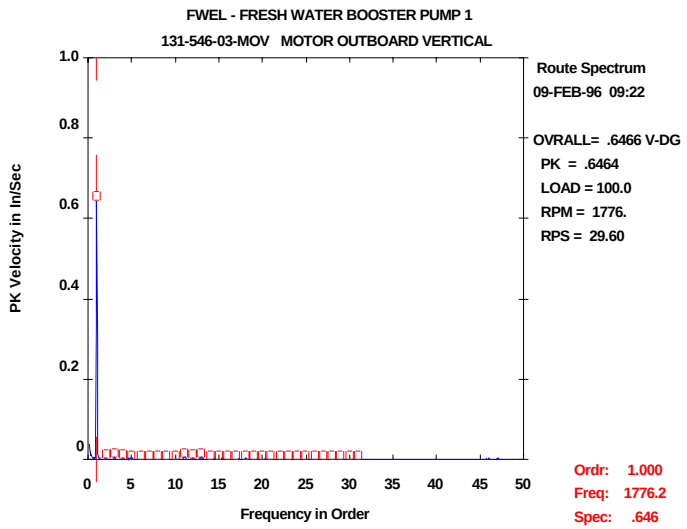
Single Spectrum - Amplitude Relations

The single spectrum plot below taken from the top of the motor, MOV, shows the predominant signal at 1X RPM (1776).

The harmonic cursor shows the lack of any harmonic activity. The sharpness of the peak indicates that it has been created from a waveform dominated by a single frequency.



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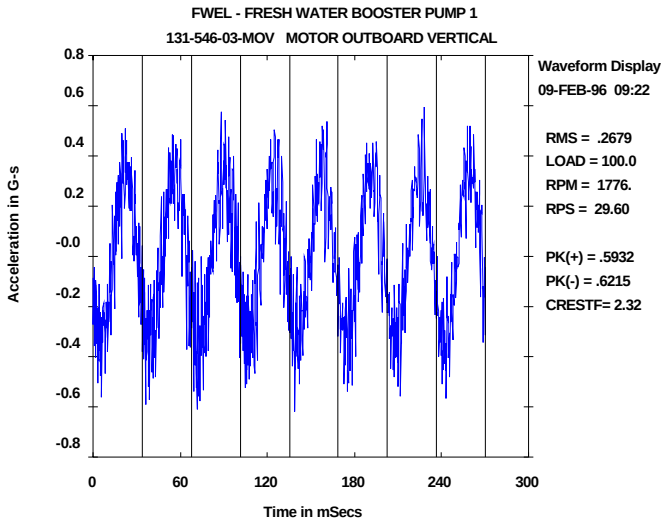
Time Waveform - Sinusoidal

The time waveform below shows the last time block stored by the analyzer.

Approximately 270 milliseconds of time (8 shaft revolutions) shows the clear one per revolution signal generated by the unbalance condition. The repeatability of the signal and the lack of impacting confirm the diagnosis.



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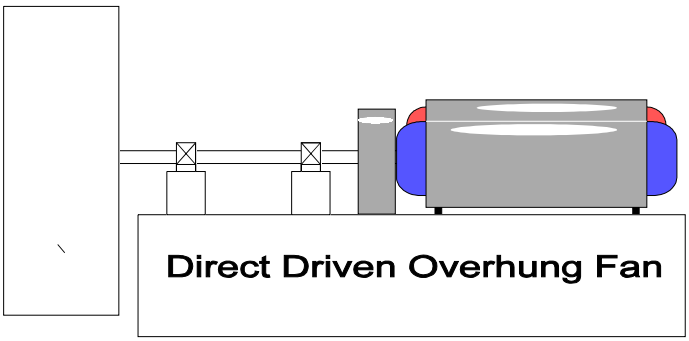
Fan Bearing Looseness Example

The fan bearing looseness data provides the initial spectral data for diagnostics and the waveform data to confirm the looseness fault diagnosis.



The fan ran in an out of balance condition for two years. The bearings now have excess clearance, allowing the shaft to move around.

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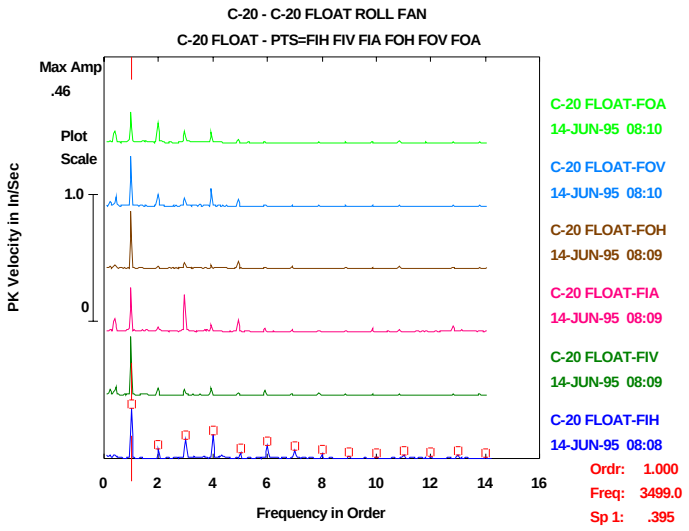
Multispectra - Data Comparison

The order-based plot below shows radial and axial measurements on the outboard and inboard turbine bearings.

The major peaks in the radial directions are all found at one order of running speed (i.e., 1 x running speed). Also, the highest amplitudes of the plot appear to be in the horizontal planes at the inboard and outboard locations.

Also note the small amounts of harmonic activity and axial data amplitude.



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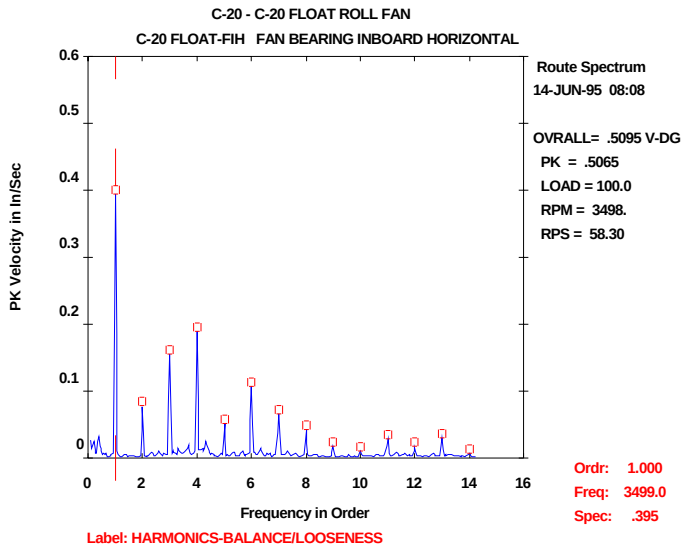
Single Spectrum - Amplitude Relations



The spectral plot above shows vibration in the horizontal direction on the fan outboard bearing.

The cursor markers note the locations of harmonics of running speed.

Virtually all the vibration energy in this spectrum is caused by turning speed and harmonics. The sides, or skirts, of this peak are also very narrow.

The number of harmonics tells us that the spectrum is derived from a complex, repetitive time waveform.

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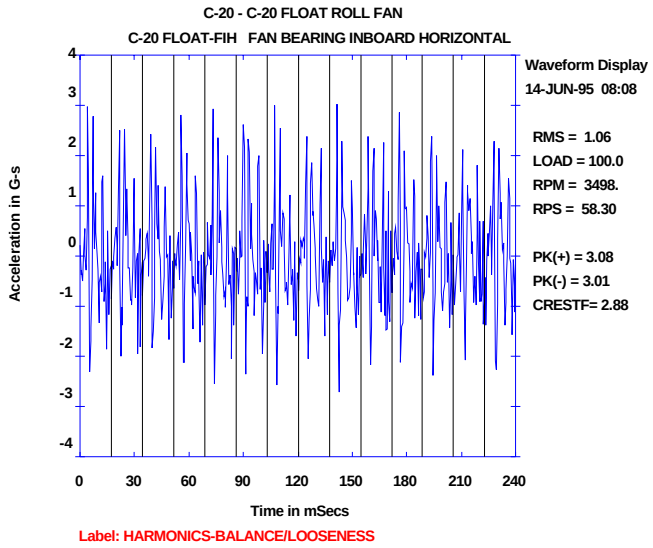
Time Waveform - Sinusoidal Character

The waveform below shows instantaneous acceleration over 240 milliseconds in the horizontal direction on the fan inboard bearing. Vertical lines show the time required for the fan shaft to make one complete revolution. The waveform is in units of acceleration since the probe used to collect data was an accelerometer.



A clear and repeatable waveform occurs once per shaft revolution, 1 x RPM. There is also multiple peaks within one revolution The waveform shows the acceleration created on the bearing housing by the looseness. The repeatability of the waveform in time with respect to the shaft turning speed and amplitude means that the vibration force is tied to the shaft running speed. This is true of waveforms collected in acceleration, since they can be influenced by high-frequency energy bursts, such as impacts.

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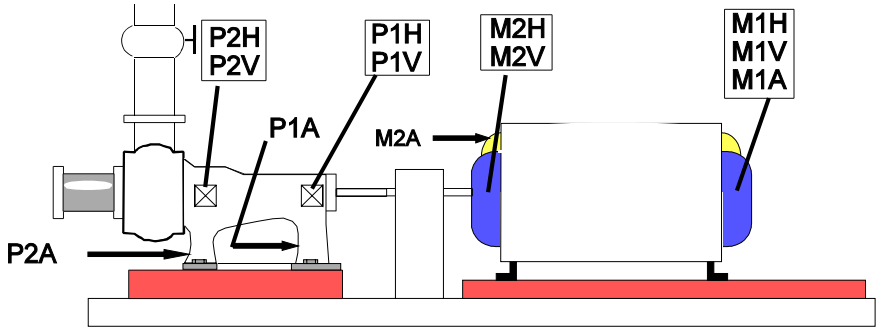
Motor to Pump Misalignment Example



The pump is installed below ground and supplies water for household use from a water treatment facility. The pump has had high vibration since installation and numerous seal/packing and bearing failures. The maintenance personnel stated that the alignment was “difficult” because the base was drilled incorrectly at the manufacturers facility.

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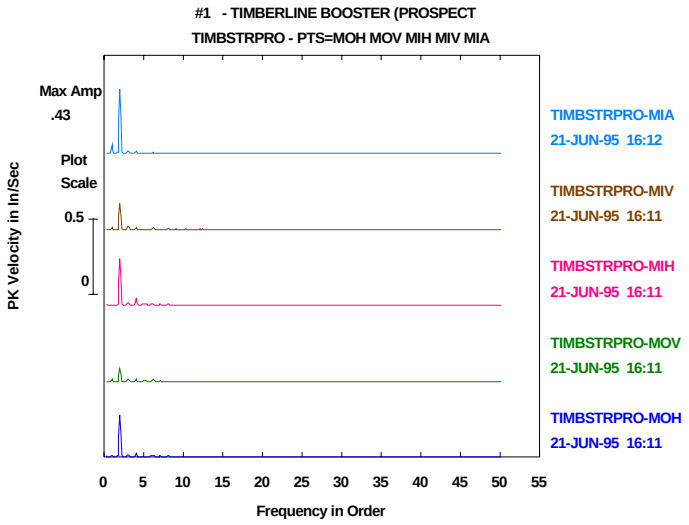
Multispectra - Amplitude Comparison



VA2CHSTY.DAT, MIS-MISALIGNMENT, TIMBSTRPRO, TIMBERLINE BOOSTER

The multispectral plot above shows inboard and outboard motor points. At first glance, the problem might appear to be unbalance. If we take a closer look we see that 2X running speed peaks are present in all directions, which may indicate an alignment or looseness problem. This is a good time to examine the time waveform.

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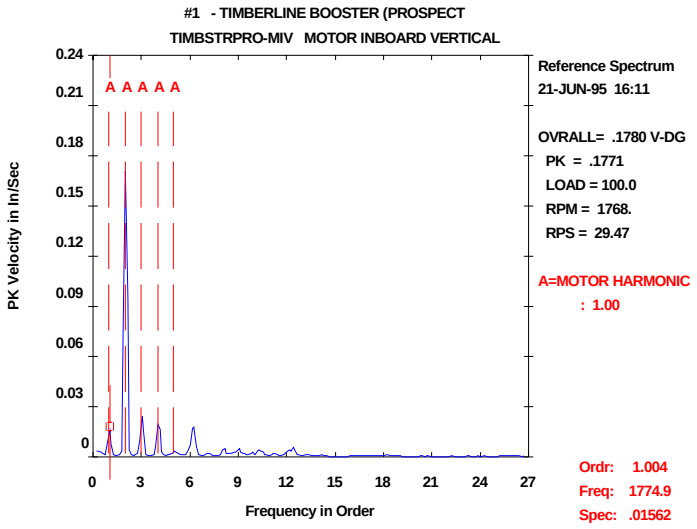
Single Spectrum - 2X T.S.



The single spectrum display of the motor inboard vertical position is shown above.

Harmonics of running speed are denoted by the fault frequency markers (dashed lines).

The first through sixth orders of running speed are visible with the 2X T.S. predominant. The peaks are sharp and very little non-synchronous energy is present, therefore the waveform is repeatable.

The diagnosis can be confirmed by viewing the time waveform.

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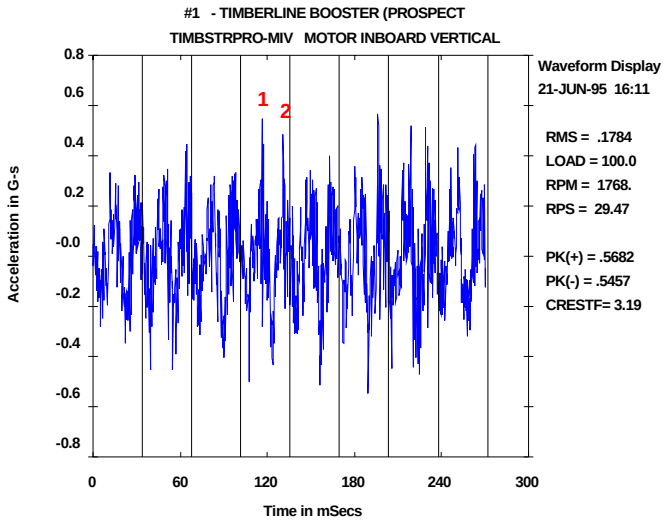
Time Waveform - Twice per Revolution



The time waveform for the motor inboard horizontal position is shown above. The vertical lines denote the time required for one shaft revolution.

The waveform is repetitive for each revolution with two distinct peaks for each period.

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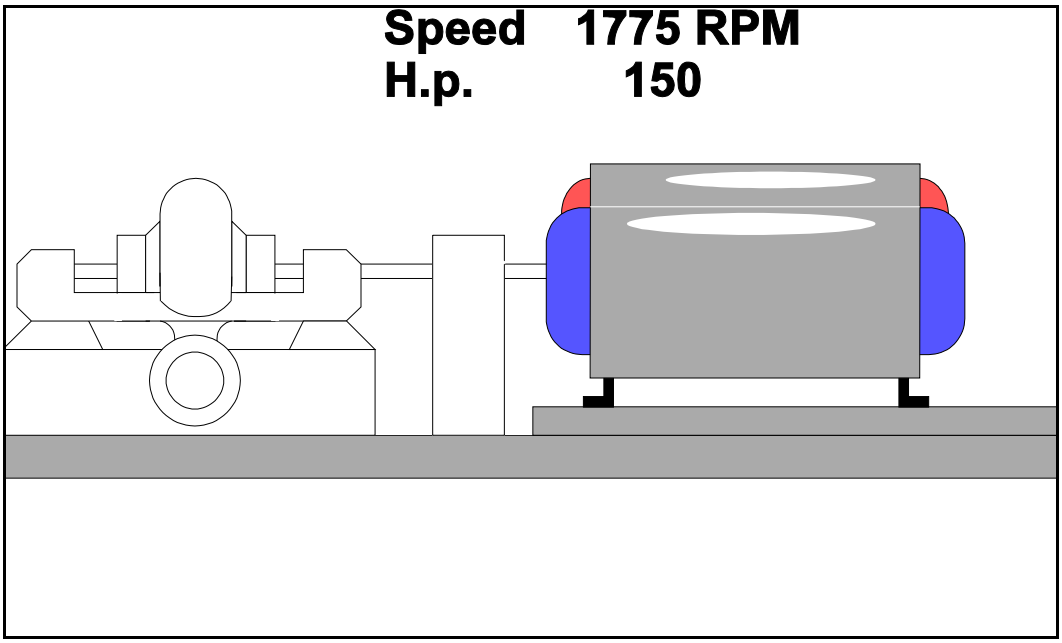
Pump Bearing Looseness Example



The diagram above shows a centerhung pump with bearing housing dimensions worn oversize .

The motor turns at about 1,775 RPM.

The worn housings makes the pump very loose . Typical of many looseness problems, this has grown worse over time. A small dimension problem has gradually made itself worse.

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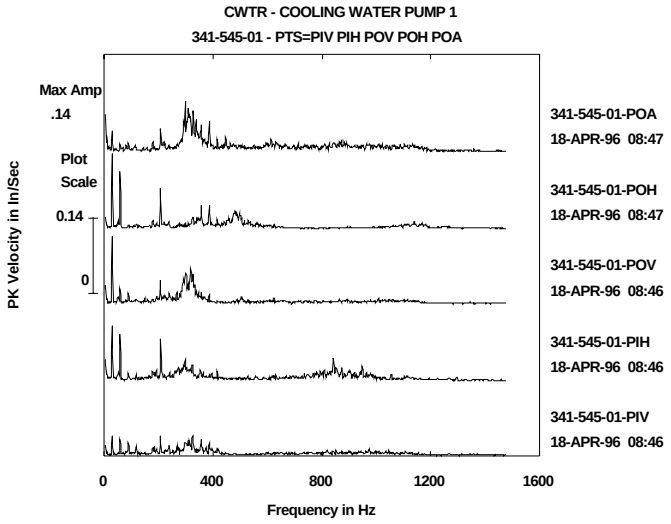
Multi spectral - Broadband



VA2CHSTY.DAT, BRG-BEARINGS, 341-545-01COOLING WATER PUMP 1

A Multi spectral plot of all five pump points is shown above.

Inboard and outboard radial measurements and an outboard axial measurement were made.

Many harmonics of running speed are visible on all measurement positions. Baseline or floor energy is also very visible.

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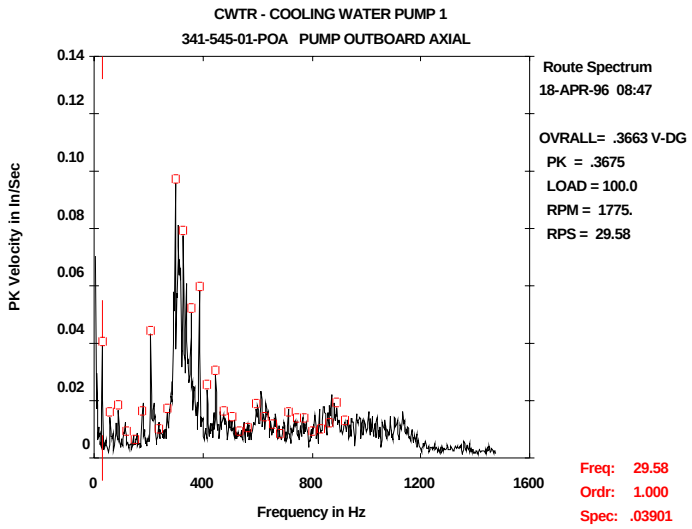
Single Spectral - 9-15x T.S. and Broadband



A full-screen view of the motor inboard horizontal position is shown above.

A cursor is positioned at 1x running speed and on the harmonics of running speed.

The peaks are broad and have wide skirts.

Notice, no individual peak exceeds .1 in/sec, but the overall energy is .3663 in/sec.

This is common with looseness. Broad humps of energy show up in the 9X to 15x running speed range.

This indicates that the time waveform cannot be cleanly transformed into a spectrum. Therefore, the waveform must have random, non-periodic energy present.

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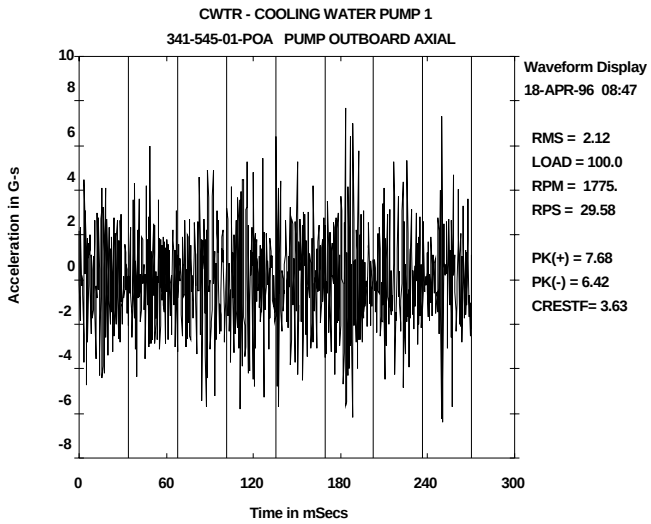
Time Waveform - Random Energy



The time waveform used to calculate the motor inboard horizontal position is shown above.

Vertical lines show the time required for the shaft to complete one full revolution. This waveform is non-repeatable and non-periodic.

There is no similarity in its pattern from revolution to revolution. Therefore, the vibration is not tied to the turning speed of the shaft. It makes sense that the waveform trace is not repeatable if the shaft does have a looseness problem and that the pattern would not be tied to the turning speed of the shaft.

Non-periodic, random patterns do not convert well in the FFT process. It is very difficult to assign specific frequencies and amplitudes to patterns in waveforms like this one.

This difficulty leads to the broadband energy humps in the spectrum. Broader humps indicate more random energy. Higher humps indicate more impacting in the waveform.

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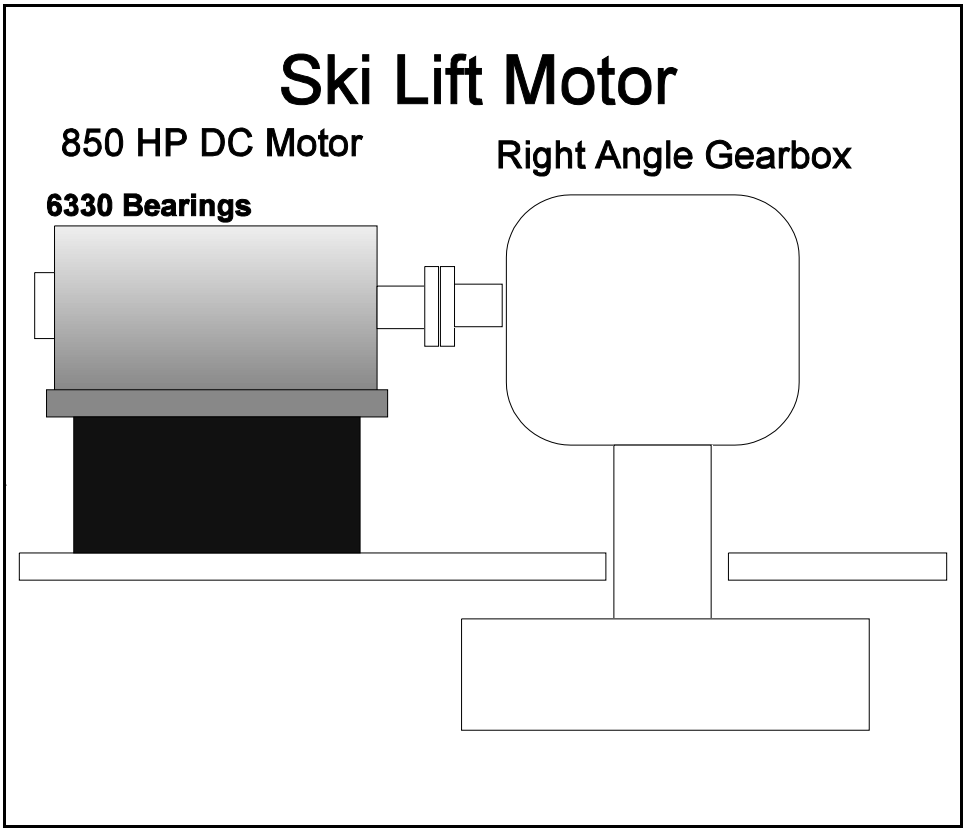
Rolling Element Bearing Example



The motor and gearbox above turn the cable reel for a large ski-lift.

The maintenance personnel reported vibration from the back end of the motor after only 200 hours operating time on a newly installed drive.

The analyst investigated and found visible flakes of a bronze colored material near the back end of the motor. The motor manufacturer was contacted to determine if the 6330 bearings had a bronze retainer and the reply given was no.

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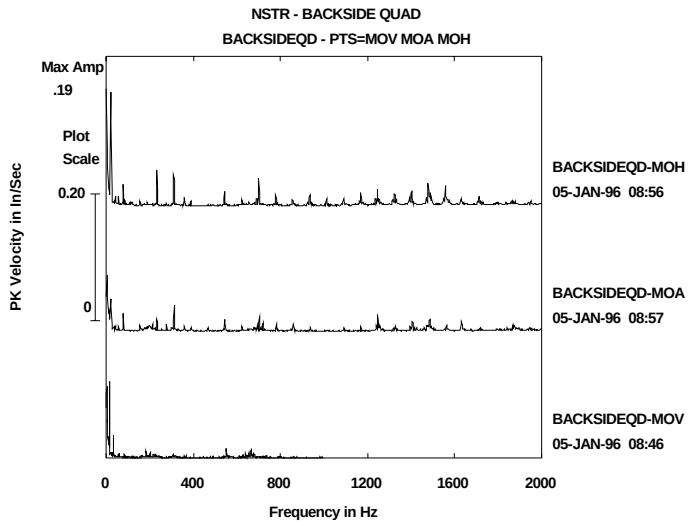
Multi spectra - Non-Synchronous Energy



VA2CHSTY.DAT, BRG-BEARINGS, BACKSIDEQD-BACKSIDE QUAD

This is a Multi spectral plot of the motor point outboard points.

All the levels appear very low in amplitude, but notice the location of the dominant peaks.

There appears to be groups of many peaks closely spaced in the mid to higher frequency range. These “mounds of energy” can indicate bearing defects.

We need to look at the spectrum a little closer.

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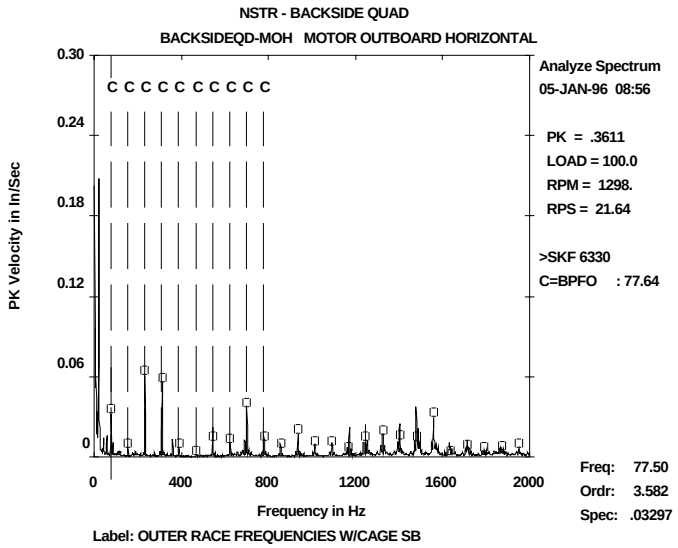
Single Spectrum - Bad Bearing



A full screen plot of the motor outboard horizontal position is shown above.

The fault frequencies for the 6330 bearing ball pass frequency outer race are marked. Notice the number of peaks surrounding the higher frequency defect harmonics.

The large number of harmonics and sidebands will be created from a complex waveform.

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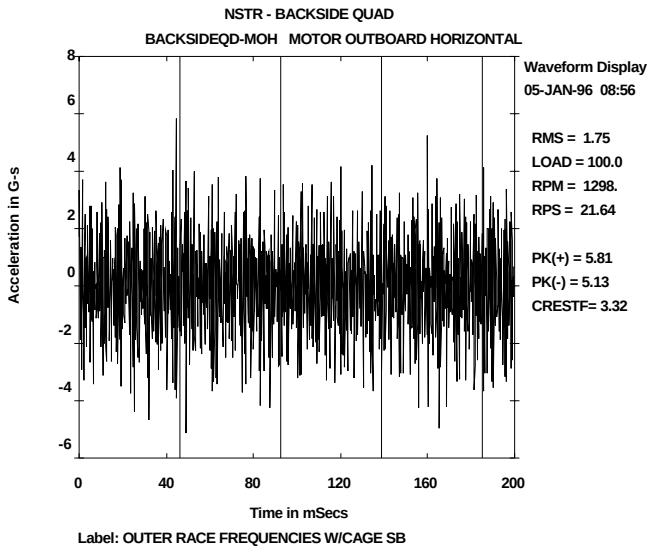
Time Waveform - Bad Bearing



The number and height of the spikes in the time waveform confirm the presence of severe impacting.

Many exceed ± 10 g's in amplitude. Remember that this is a very large DC motor with an armature that weighs 4,000 lbs., so large amounts of energy are generated by these impact levels. This impact energy is absorbed by the bearings and is damaging them rapidly.

The waveform shape is random and complex. This shape cannot be transformed into a clean spectrum, so the spectrum on the previous page with broad humps of energy is created.

The motor was disassembled in place because of the difficulty of removing a large motor in 12 feet of snow at 11,000 foot elevation. The bearing cage turned out to be bronze! It was deteriorating and did not have much life left. The outer race had major spalls from impacting balls. The bearing was replaced.

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Summary

Waveform data may be used for much more than what is typically seen in industry. The ability to check for specific characteristics such as periodicity and modulation, helps the analysis process.

Energy balance (asymmetry) may be checked for direction of signal and for the predominant traits of the signal.

Overall waveform is much more understandable and useful than most would lead us to believe. However, this section enhances your analysis abilities using the time waveform.

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Section Eleven
Peak / Phase Analysis

Objectives

- Define peak/phase data and the convention used to measure it.
- Describe reasons for monitoring peak/phase data.
- Describe and interpret display formats for peak/phase data.

General Description

The monitor peak/phase data defines the location (in degrees) of a machine's vibration peak with respect to a fixed reference mark on the rotor. Several reasons to collect peak/phase data appear below:

- Peak/Phase data provides information regarding resonant frequencies of a machine when collected during startup or coastdown.
- Peak/Phase data proves useful for field balancing work.
- Peak/Phase data helps diagnose various machinery faults such as coupling misalignment and bent shafts.

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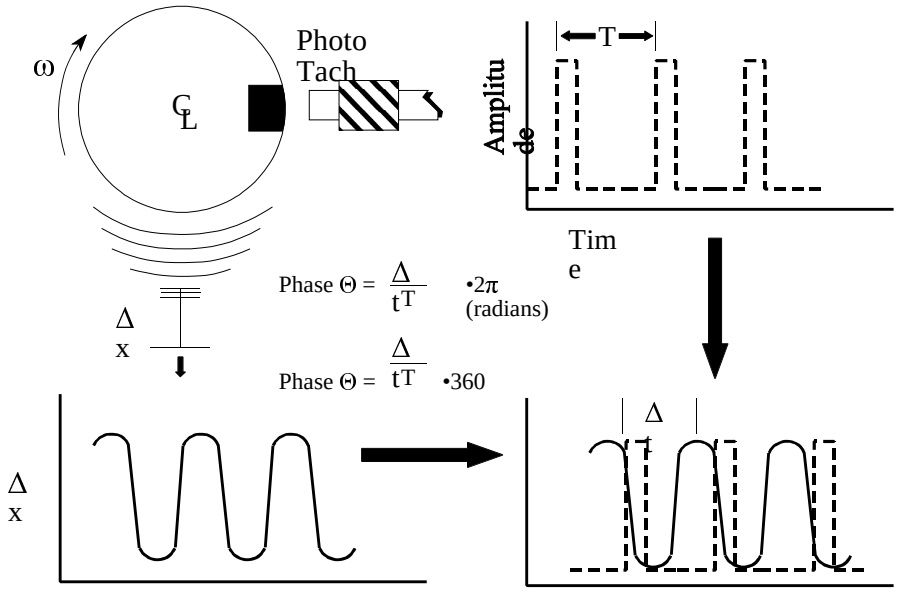
Phase Data

Phase can be defined in two ways. First, ***phase*** is that part of a cycle (0° to 360°) through which a particular part of a machine travels relative to a fixed reference point. Second, phase refers to the location in degrees that marks a machine's high vibration peak and its frequency relative to a fixed reference mark on a rotating component of the machine. This second type of phase is also known as ***synchronous phase.***



Single Channel Phase Measurement (@ 1XTS)

Phase varies with the monitored frequency. For machinery diagnostic work, synchronous phase refers to 1xTS or any harmonic of turning speed up to 128 orders. CSI measures phase counter to the rotation of the shaft.

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Specific Fault Types

Unbalance

Static unbalance shows a zero degree phase shift across the rotor radial-to-radial or horizontal-to-horizontal, a 90° (±20°) phase shift from vertical to horizontal at the same bearing location.

Dynamic unbalance shows a phase shift across the rotor radial-to-radial or horizontal-to-horizontal related to the heavy spots on each end of the rotor. If the heavy spots are 180° out of phase on each end, then the phase measurements will also be 180° out of phase.

Misalignment

Angular misalignment typically shows a 180° (± 30°) phase shift across the coupling in the axial direction.

Parallel misalignment tends to show a 180°(± 30°) phase shift across the coupling in a radial direction.

Bearing misalignment (cocked bearing) shows a 180°(± 30°) phase shift from one side of the bearing to the other or from the top of the bearing to the bottom.

Looseness and Soft Foot

Phase reading with looseness will be erratic from point to point around the machine train. A soft or loose mounting foot usually shows a phase shift from the foot itself to the foundation. This shift is often greater than 90°. Also, a soft foot shows a phase difference from the other machine feet.

Resonance

Typically, for a lightly damped system, the phase shifts 180° as the turning speed moves through the resonant frequency with approximately a 90° phase shift occurring at the resonant frequency.

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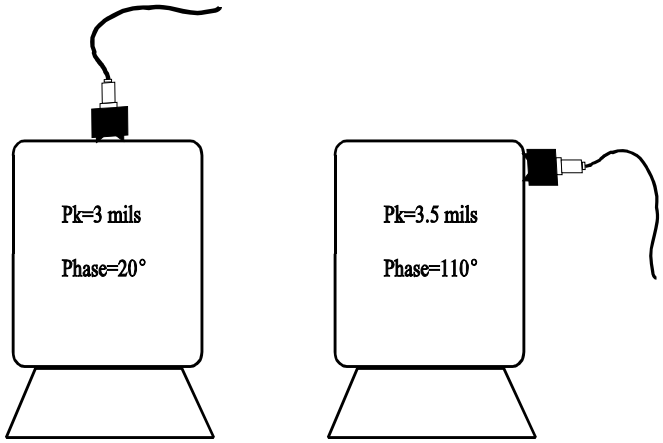
Phase Measurements

Phase is measured by using the once-per-revolution pulse as a reference point. Depending on the machine fault, a machine's structure moves either as one unit, ***in phase***, or separately, ***out of phase***.

Usually the amplitude and the phase of the 1xTS frequency is of the most interest. The relationship between the high vibration amplitude and the tachometer pulse determines the phase. ***Phase*** is the difference in time, measured in degrees, between the leading edge of the tach pulse and the peak amplitude in the waveform. As the vibration sensor changes from a vertical position to a horizontal position, a 90° shift, the time between the tach pulse and the peak amplitude will also change 90°.



Transducer shifted 90° yields a 90° shift in phase.

Monitoring the amplitude and phase response during a startup or coast down is a method of using peak and phase measurements to determine resonances. The analyzer should be placed in monitor peak and phase mode at the turning speed harmonic of interest. The machine should then be shut off. Take care to ensure that the peak and phase data are sampled fast enough to capture meaningful data.

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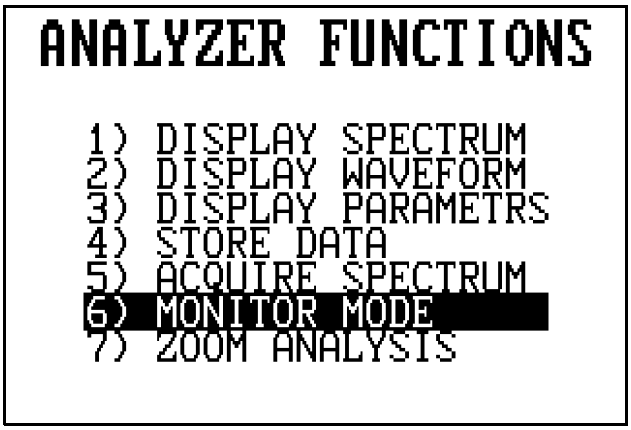
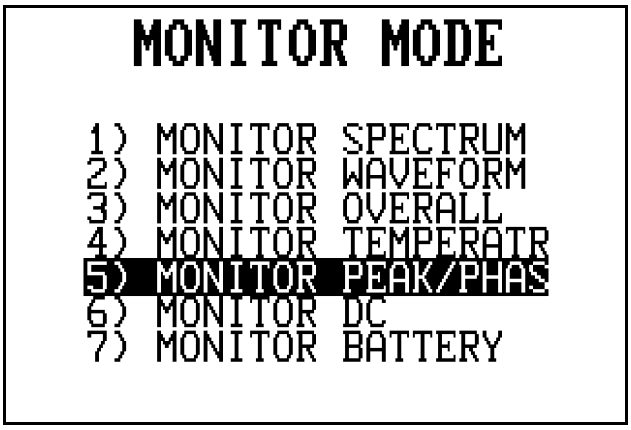
Monitor (Synchronous) Peak/Phase Lab

This lab covers hands-on operation of the Model 2120 Machinery Analyzer using the Monitor Peak/Phase option.



Press the **Analyze** key to display the Analyzer Functions menu. Highlight **6) MONITOR MODE** and press **Enter.**



Highlight **5) MONITOR PEAK/PHASE** and press **Enter.**

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```
MONITOR PEAK/PHASE

ORDER (XRPM)    :1
BANDWIDTH(XRPM):0.10
AVERAGER ENABLD: NO
MINIMUM RPM     :0
TACHMTR TIMEOUT: YES
FS RANGE        :10.00

NO DATA STORED
```

For **ORDER (XRPM)**, specify an integer multiple of the machine turning speed or RPM, **1** is normally used.

BANDWIDTH (XRPM) sets the bandwidth of the tracking filter. The analyzer attenuates all frequency components outside the specified bandwidth. Multiply peak frequency times parameter bandwidth for filter bandwidth. Accepted values range from 0.05 to 0.5, but **0.10** works well for most applications. A more narrow filter (a smaller bandwidth) helps isolate the indicated peak, but it requires more time per measurement.

When you set **AVERAGER ENABLED** to **YES**, the analyzer calculates a new vector average for each peak/phase average. This helps when readings fluctuate or when nearby components modulate the frequency. Use the **CLR** key at any time to clear the averaged vector and restart the averaging process. The **TACHMTR TIMEOUT** will display the **NO TACH** message when set to **YES**. Setting this to **NO** will disable the message. The **FS RANGE** when set to **0**, will AutoRange the signal before each measurement. Setting a value will allow you to collect data more quickly. The units are in the same units as the waveform. Press **Enter** to collect data.

```
MONITOR PEAK/PHASE

RPM  :  1775.
PEAK :  0.555  IN/SEC
PHASE:    263 DEGREES


[PRESS CLEAR TO CLEAR DATA]
[PRESS ENTER TO EXIT SCAN ]
```

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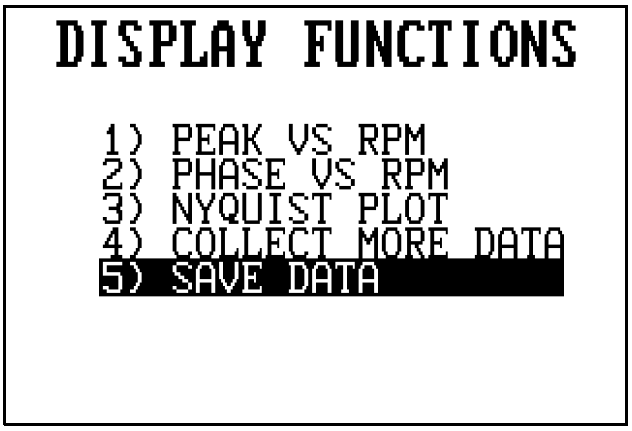
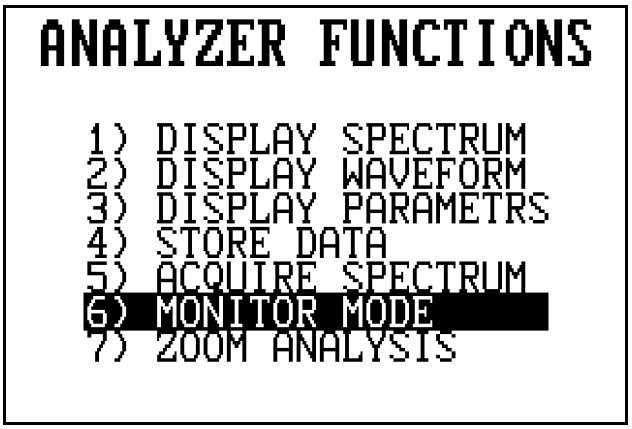


When you finish data collection, press **Enter** to show the Display Functions menu. Highlight **5) SAVE DATA** and press **Enter.**

Coastdown / StartUp Tests



To view the collected data, follow the same procedure used to monitor the 1xTS peak and phase data. Press the **Analyze** key. Highlight **6) MONITOR MODE** and press **Enter** to continue.

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```
MONITOR MODE

1) MONITOR SPECTRUM
2) MONITOR WAVEFORM
3) MONITOR OVERALL
4) MONITOR TEMPERATR
5) MONITOR PEAK/PHAS
6) MONITOR DC
7) MONITOR BATTERY
```

Highlight **5) MONITOR PEAK/PHASE** and press **Enter.**

```
MONITOR PEAK/PHASE

ORDER (XRPM)    :1
BANDWIDTH(XRPM):0.10
AVERAGER ENABLD: NO
MINIMUM RPM     :0
TACHMTR TIMEOUT: YES
FS RANGE        :10.00

NO DATA STORED
```

The setup above is identical to the previous setup.

```
MONITOR PEAK/PHASE

RPM  :  1775.
PEAK :  0.555  IN/SEC
PHASE:    263 DEGREES

[PRESS CLEAR TO CLEAR DATA]
[PRESS ENTER TO EXIT SCAN ]
```

With the machine running, the Monitor Peak/Phase menu displays the amplitude and phase data on the machine before the machine is shut off.

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```
MONITOR PEAK/PHASE

RPM  :  1241.
PEAK :  0.913  IN/SEC
PHASE:    218 DEGREES

[PRESS CLEAR TO CLEAR DATA]
[PRESS ENTER TO EXIT SCAN ]
```

You can monitor the amplitude and phase as the machine coasts down to a stop as the power to the machine is shut off.

```
MONITOR PEAK/PHASE

RPM  :  932.3
PEAK :   2.08  IN/SEC
PHASE:    103 DEGREES

[PRESS CLEAR TO CLEAR DATA]
[PRESS ENTER TO EXIT SCAN ]
```

The machine is near resonance at this point. Notice how the amplitude has reached over 2 In/Sec and the phase has shifted 115 degrees.

```
MONITOR PEAK/PHASE

RPM  :  751.0
PEAK : 0.0383  IN/SEC
PHASE:     59 DEGREES

[PRESS CLEAR TO CLEAR DATA]
[PRESS ENTER TO EXIT SCAN ]
```

As the machine coasts to a stop, the amplitude is negligible and the phase has shifted more. Press **Enter** to exit the data scan.

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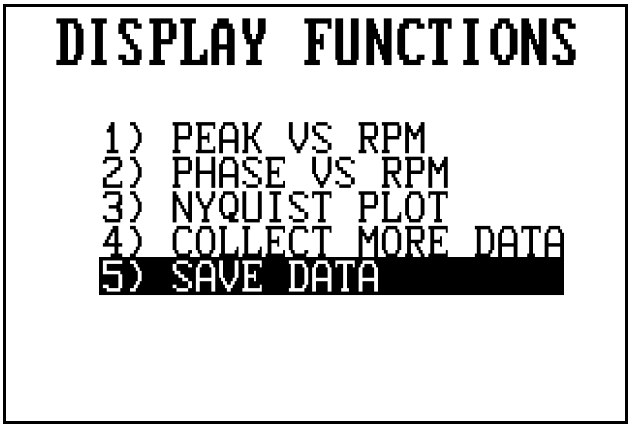
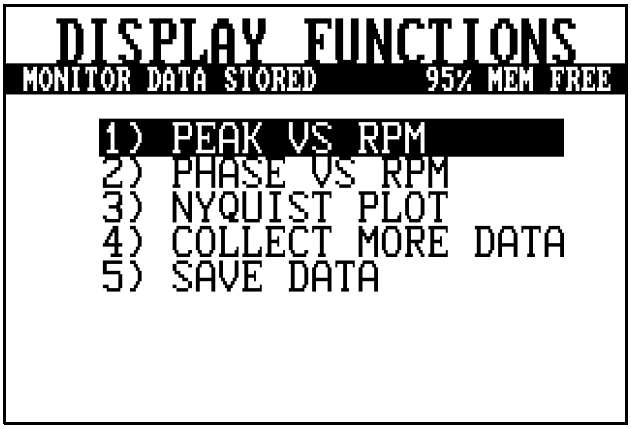
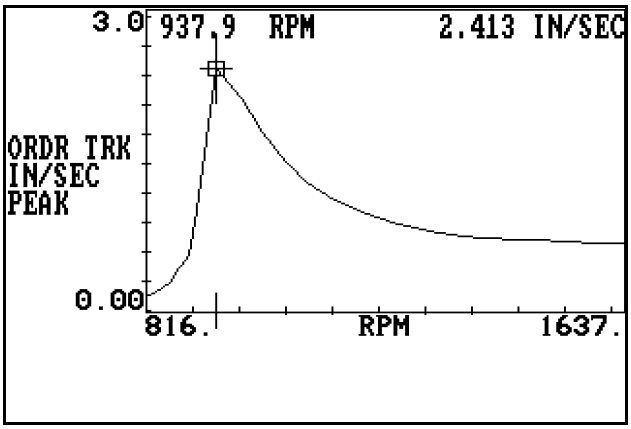


To avoid inadvertently losing data, it is best to store the data right after data has been collected.



After data is stored, view the Peak vs. RPM data by selecting **1) PEAK VS RPM** and pressing **Enter.**



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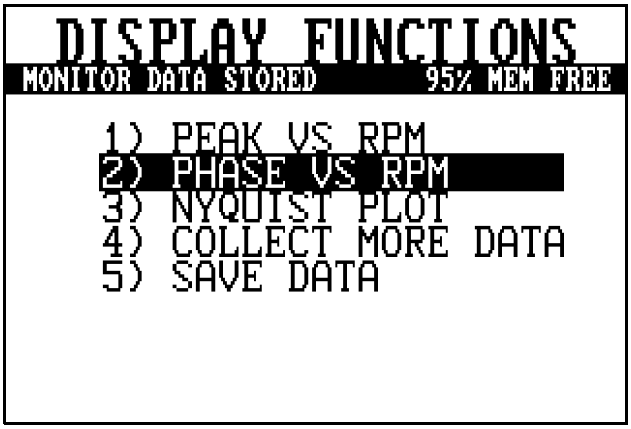
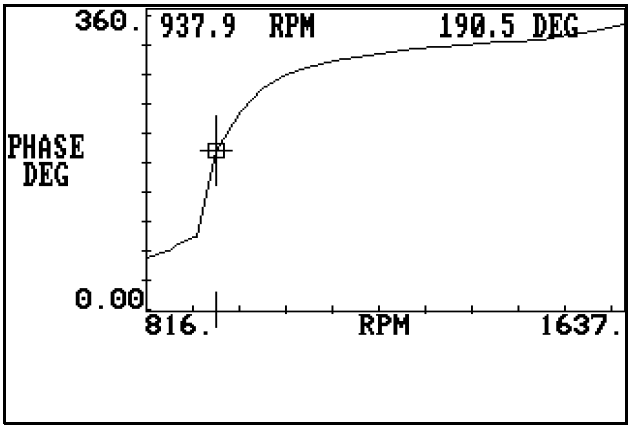
The resonance appears in the data at a turning speed of 937.9 RPM.

Press **Enter** to return to the Display Functions Menu.



Select **2) PHASE VS RPM** to view the phase change with respect to speed.



The phase has gone through about a 180° phase shift with about a 90° phase shift at resonance.

This plot helps verify whether the peak you observed was caused by resonance. Press **Enter** to return to the Display Functions menu.

The phase reading from a monitored machine should change by approximately 180° as it runs up to speed, passes through the resonance range(s), and moves into the free space region.

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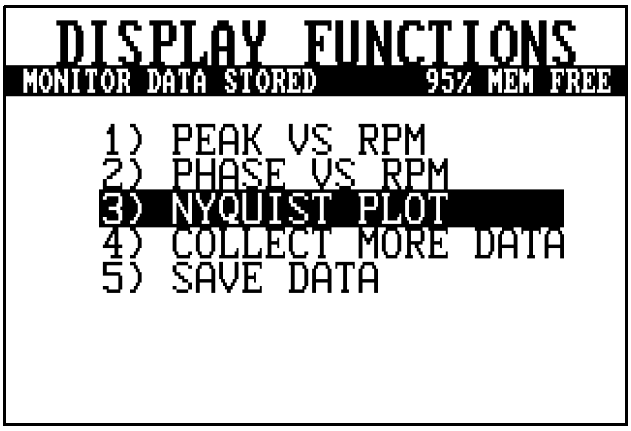
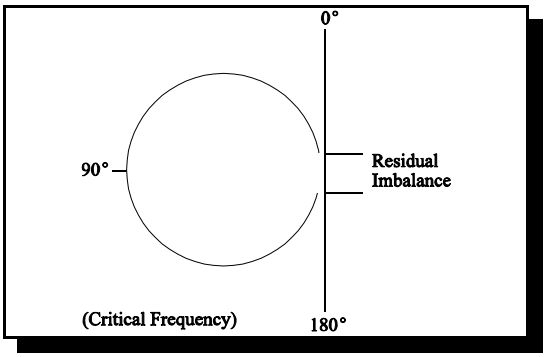
The phase reading always becomes unstable at the resonant frequency. Phase at resonance should, however, differ by roughly 90° from measurements in the free space region. The same holds true for measurements taken when the machine turns too slow to experience vibration amplification from resonance.



Highlight **3) NYQUIST PLOT** and press **Enter** to view a polar plot of the data. Bode plots (Peak or Phase vs. RPM) yield important information about resonance. The presence of runout or a bow in the shaft, however, can significantly alter the appearance of the plots. Nyquist plots, on the other hand, remain unaffected by runout and bowed shafts. Use nyquist plots to confirm any conclusion based on Bode plots.

The illustration below shows the important features of the Nyquist plot. Note that any loops in the plot are evidence of the other resonant frequencies.



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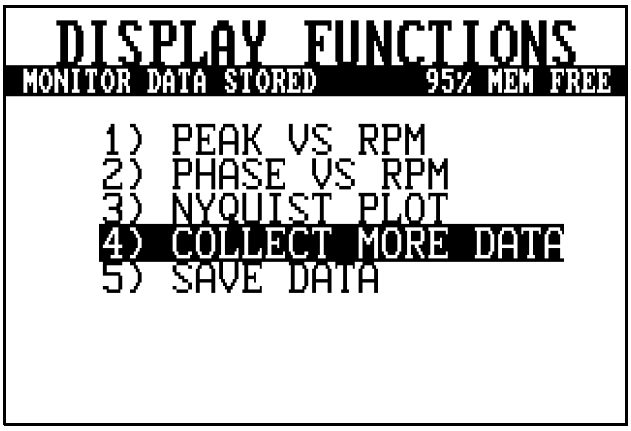

If the machine happens to be running and you wish to collect more data, press **4) COLLECT MORE DATA**. If you collect more data, you may want to clear the data before continuing.

Interpreting the Coastdown Results

Resonant frequencies on a particular machine do not change unless degradation occurs (e.g., cracked welds or eroded grout). For critical machines, know where resonances occur before repair or rebuild work begins. This information provides a baseline for comparison after the work is finished.

Resonant frequencies should remain the same before and after a rebuild unless you make structural or component modifications. If the before and after frequencies do not coincide, most likely the machine was not assembled properly. Loose, missing, or improperly fitted components can cause resonant frequencies to shift.

Machines that experience frequent bearing failures or structural cracking often operate at or near a resonant frequency. Use coastdown analysis to verify this condition.

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Viewing Stored Peak and Phase Data

```
ANALYZER FUNCTIONS

1) DISPLAY SPECTRUM
2) DISPLAY WAVEFORM
3) DISPLAY PARAMETRS
4) STORE DATA
5) ACQUIRE SPECTRUM
6) MONITOR MODE
7) ZOOM ANALYSIS
```

Press the **Analyze** key and highlight **6) MONITOR MODE** and press **Enter.**

```
MONITOR MODE

1) MONITOR SPECTRUM
2) MONITOR WAVEFORM
3) MONITOR OVERALL
4) MONITOR TEMPERATR
5) MONITOR PEAK/PHAS
6) MONITOR DC
7) MONITOR BATTERY
```

Select item **5) MONITOR PEAK/PHASE** and press **Enter.**

```
MONITOR PEAK/PHASE

ORDER (XRPM)    :1
BANDWIDTH(XRPM):0.10
AVERAGER ENABLD: NO
MINIMUM RPM     :0
TACHMTR TIMEOUT: YES
FS RANGE        :10.00

NO DATA STORED
```


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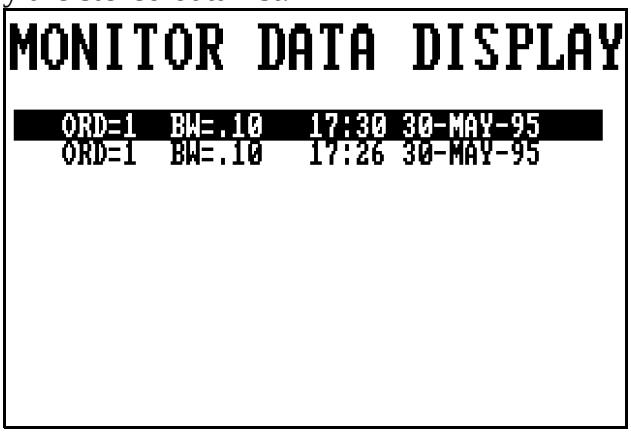
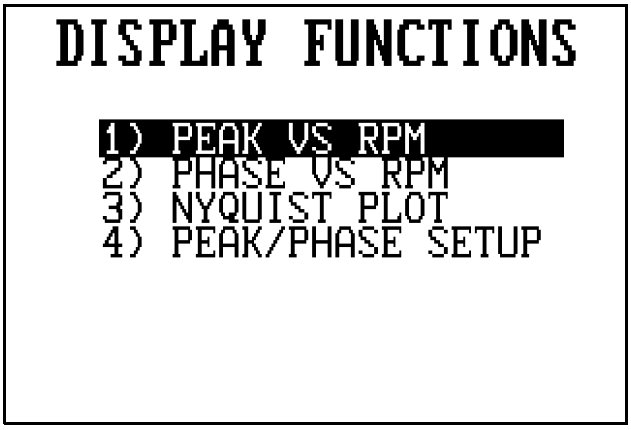
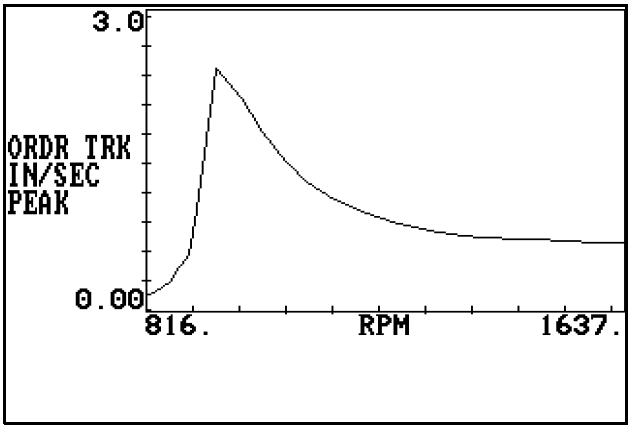
Press **F6** to display the stored data list.



Highlight the data set of interest and press **Enter** to display the data.



Select the desired plot and display function. In this case, select **1) PEAK VS. RPM.**



The peak versus RPM data displays here.

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Summary

It is important to be able to monitor the amplitude and phase data at different multiples of turning speed. This is another tool for doing “investigative” vibration analysis. Phase is a useful tool for analyzing many of the common faults in machinery. It also helps the analyst to visualize the operating movement to diagnose the root cause of the vibration.

Monitoring the amplitude and phase data during a coastdown or a startup test is a powerful analysis tool for confirming the presence of a resonance. Remember to look for the amplitude increase relative to the frequency as well as the phase change relative to the frequency.

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Section Twelve
Gears

Objectives

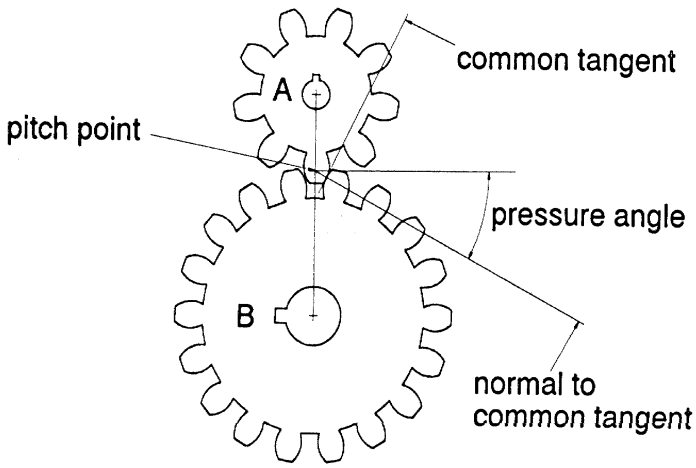
- Describe the basic purpose of gears and their components with needed gear theory.
- Identify transmission error and other gear defects and the effect on vibration data.
- Select the best sensor, sensor location, and measurement point setup for meaningful data collection.

Gear Design

Gears are commonly used in industry to provide the speed and power transmission requirements needed in industrial applications.

Gears can provide these speed changes and torque transmission without slip.

Gear designs have specific characteristics that can affect the measured vibration. Gears are often considered too complex to diagnose their defects properly, but with an understanding of a few gear design basics and terminology, troubleshooting gearboxes can be accomplished more easily.



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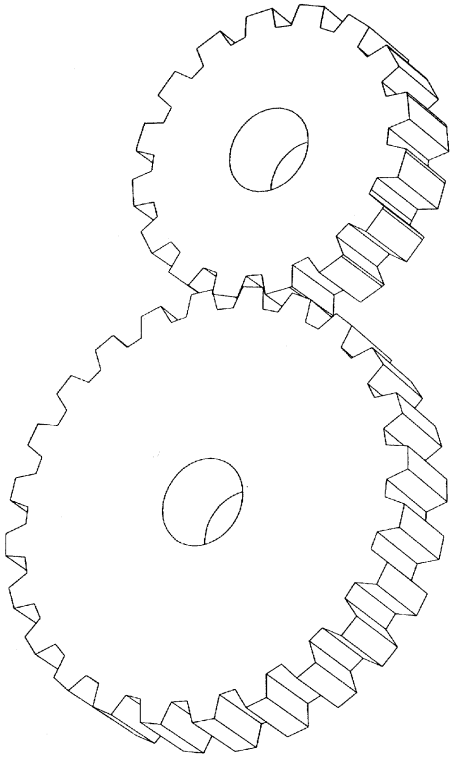
Gear Types

Different gear types are available for various speed and power considerations.

Each of these different gear types shows the same basic vibration patterns when gear defects appear.

Spur Gears



Spur gears are most commonly thought of when discussing gears. The teeth are cut parallel to the shaft. These gears are good at power transmission and speed changes, but are noisier than other gear types.

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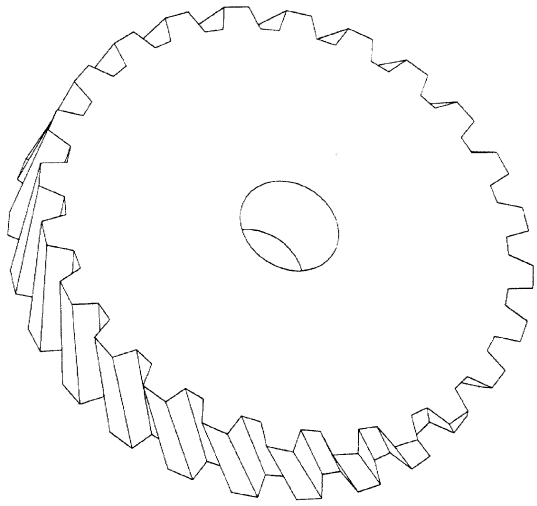
Helical Gears



Helical gears have teeth cut at an angle to the shaft.

These gears are much quieter than spur gears; but, due to the angular nature of the gear meshing, axial thrust and, therefore, **axial vibration is higher** on these gears than spur gears.

To avoid the higher axial thrust, double helical gears are used. These gears, often called herringbone gears, are divided in the middle with each side having an equal magnitude angle but in an opposite direction.

If a gap exists between the two halves of the gear, then it is a ***double helical gear.***

If no gap exists and each tooth is continuous, then the gear is called a herringbone gear.

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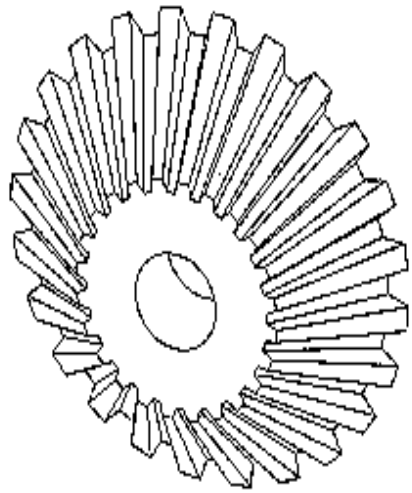
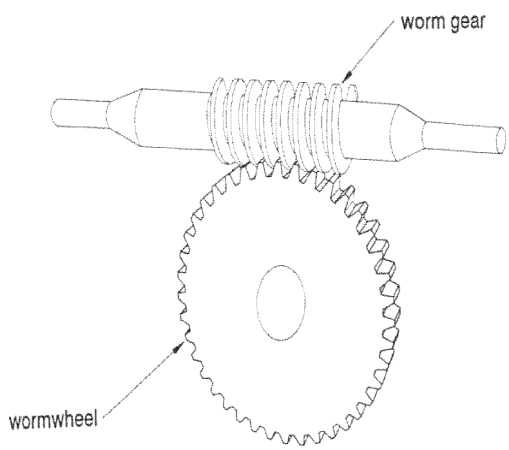
Bevel Gear



Right angle gearboxes transmit power to an output shaft perpendicular to the drive shaft. These gears may use a ***bevel gear*** design to transmit the power better. The bevel gear may have a straight bevel or a spiral bevel.

Worm Gear



$$\textit{(For Worm Drives)} \qquad GMF = (\#T)_{out} * (TS)_{out}$$

A ***worm gear*** is also used to transmit the rotational motion between perpendicular shafts. The worm has one or more teeth wrapped around its shaft. It drives a worm wheel which has the appearance of a helical gear.

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Basic Gear Theory

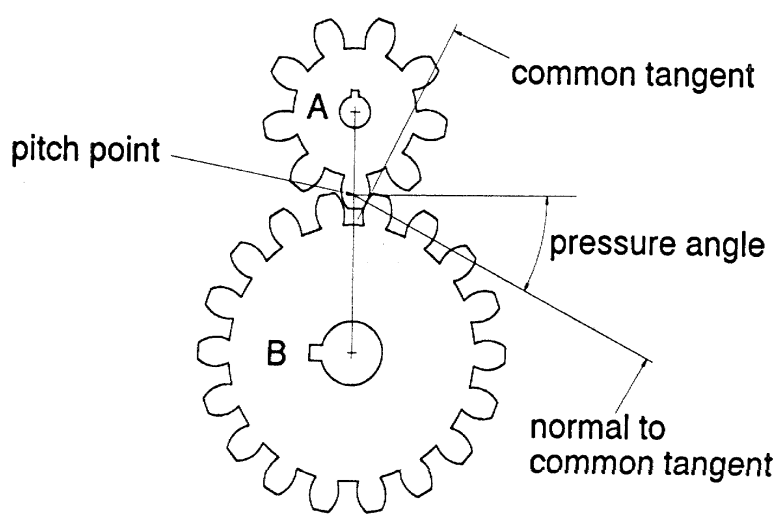
Tooth Shape

All the previously mentioned gear types use the same basic tooth design, called a tooth profile. The best tooth profile is one that allows for the radial velocity of the gears to be constant.

For modern gearing, the tooth profile that works best is called the involute. The involute design minimizes the effect of fabrication errors on the radial velocity of the gears keeping the vibration and noise levels down.

Conjugacy

The purpose of a gearbox is to provide power and or speed changes with a minimum of excessive noise and vibration. To accomplish this, the power from the drive gear must be transmitted though a line perpendicular to the common tangent and intersecting the center-to-center line. The common tangent is a line tangent to both of the meshing gears. This point of intersection is called the pitch point. The pitch point of each gear tooth must be on the center-to-center line between the gears. The circle that connects the pitch points is called the pitch circle.



This is the principle of conjugacy. The use of the involute tooth profile allows this condition to be met more easily.

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Prime Number Theory

The number of teeth on each gear can be factored down to a series of prime numbers.

Prime numbers are 1, 2, 3, 5, 7, 11, 13, 17, 19, etc. For example, the number 10 can be broken down to its prime factors 1 x 2 x 5, the number 26 can be factored into 1 x 2 x 13. Prime numbers are important when trying to understand some of the gear defects and their frequency components.

When the largest prime factor common to both gears, called the ***largest common factor*** (LCF), is 1, one tooth on a drive gear will mesh with every tooth on the driven gear before it remeshes with the first tooth on the driven gear.

If the LCF is greater than 1, some teeth will mesh more often; and this leads to an reduced gear life. Also, machining defects and wear patterns develop that show up as defect frequencies based on the largest common prime factors between the meshing gears.

Vibration Analysis

Vibration analysis of gears can provide a wealth of information about the mechanical health of the gears. This section discusses the source of the frequencies that may be present in a gear box.

The source of most of the defect frequencies is transmission error between two meshing gears.

Transmission error is caused by machining errors, tooth deflections, looseness, eccentricity, or anything that causes the power to be transferred through any point other than the pitch point.

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Gear Mesh Frequency

Gear mesh frequency (GMF) is the most commonly discussed gear frequency.

However, **GMF by itself is not a defect frequency**.

GMF is always present in the spectrum regardless of gear condition. Its amplitude may vary, however, depending on the gear condition.

Single Reduction

GMF is simply defined as the number of teeth on a gear multiplied by its turning speed.

GMF = (#Teeth) * (Turning Speed)

Example:

GMF = 44t * 29.59 Hz = 1301.96 Hz

1301.96 * 60 = 78,117.6 cpm

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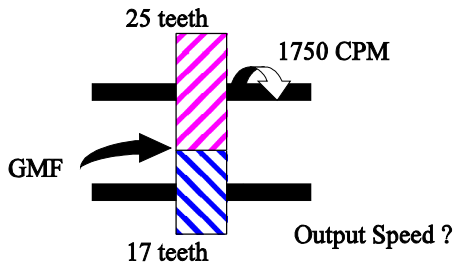
If the turning speed in the above equation is in units of RPM (or CPM), the GMF will be in units of CPM. If turning speed is in orders, then the GMF will be in orders. This relationship can be used to advantage when trying to determine the output speed of a driven gear when the input speed and the number of teeth on each gear is known. This is possible because any two meshing gears must have the same gear mesh frequency. Therefore, the above equation can be rewritten slightly.

$$(\#T)_{in} * (TS)_{in} = GMF = (\#T)_{out} * (TS)_{out}$$

When faced with the need to calculate an output speed for a single reduction gear drive.

Calculate the gear mesh frequency for the known gear and divide by the number of teeth on the output gear.

This results in the determination of the output speed. (This is also the same as multiplying the input speed by the gear reduction ratio.)



For example, if the input speed is 1750 CPM and the input gear has 25 teeth and the output gear has 17 teeth, then these values can be put into our relationship and we can find the turning speed of the output gear.

$$(\#T)_{in} * (TS)_{in} = GMF$$

$$(25) * (1750) = 43{,}750 \text{ CPM}$$

$$GMF / (\#T)_{out} = (TS)_{out}$$

$$43{,}750 / (17) = 2573.5 \text{ CPM}$$

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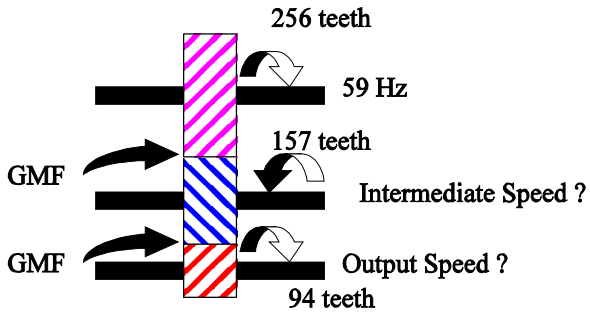
Multiple Reduction

A multiple reduction gearbox is not any more difficult to evaluate if two facts are remembered.

❶ Gear mesh frequency is the product of the number of teeth on the gear and its turning speed.

❷ Any two meshing gears must have the same gear mesh frequency.

The following example demonstrates these two principles.



The input drive gear's turning speed is 59 Hz and it has 256 teeth. It meshes with an intermediate gear that has 157 teeth and an unknown turning speed, $(TS)_{in}$.

The intermediate gear meshes with the output gear that has 94 teeth and an unknown turning speed, $(TS)_{out}$.

First determine the gear mesh frequency for the input gear.

$$(\#T)_{in} * (TS)_{in} = GMF$$

$$(256) * (59) = 15{,}104 \text{ Hz}$$

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Since the gear mesh frequency is the same for two meshing gears, the intermediate GMF is also 15,104 Hz.

The turning speed for the intermediate gear, $(TS)_{int}$, is calculated below.

$$GMF / (\#T)_{int} = (TS)_{int}$$

$$15104 / (157) = 96.2 \text{ Hz}$$

Again for the output gear, the gear mesh frequency is the same between two meshing gears. The output gear speed is determined below.

$$GMF / (\#T)_{out} = (TS)_{out}$$

$$15104 / (94) = 160.7 \text{ Hz.}$$

Even complicated gear drives can be computed if they are examined one meshing pair at a time.

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Worm Gears

Worm gears are sometimes confusing because there is a question as to how many teeth are on the input worm drive.

In the case of a worm gear, it is not the number of teeth that is of concern (often a worm drive only has one tooth), but the number of flights on the worm gear.

The flights refer to the number of teeth that mesh with the driven gear during one revolution of the worm drive.

This can be readily identified if the output gear speed, the number of teeth on the output gear and the input shaft speed are known.

In this example an output drive gear with 24 teeth turning at 10 Hz is driven by a worm gear turning at 29.5 Hz.

The number of flights (#F) on the input gear can be determined as follows.

$$(\#T)_{out} * (TS)_{out} = GMF$$

$$(24) * (10) = 240 \text{ Hz}$$

$$GMF / (TS)_{in} = (\#F)_{in}$$

$$240 / 29.5 = 8.13$$

This worm gear has 8.13 flights meshing with 24 teeth on the output gear.

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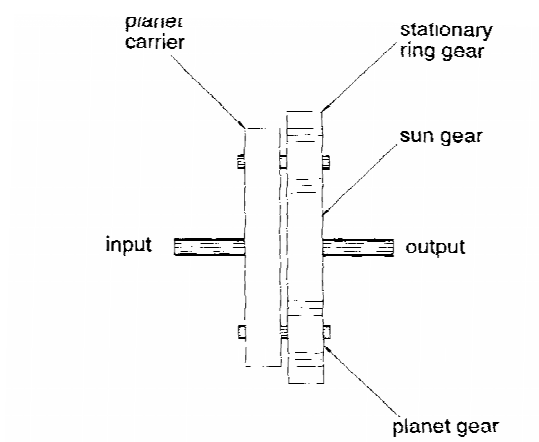
Planetary Gears

Probably the most confusing gear mesh frequency to calculate is for a planetary gear set. There are many different types of planetary gear designs. One of these is shown below. In this planetary gear, there are some new components that need to be identified.

The input shaft is attached to the planet carrier which drives the planet gears.

The planet gears mesh with the ring gear and the sun gear which drives the output shaft. For this gear set, the GMF is equal to the number of teeth on the planet gear $(\#T)_{planet}$, multiplied by the speed of the planet gear. GMF is also equal to the number of teeth on the sun gear $(\#T)_{sun}$ multiplied by the output shaft speed.



Now, the speed of the planet $(TS)_{planet}$ is determined by multiplying input shaft speed $(TS)_{in}$ by the number of teeth on the ring gear $(\#T)_{ring}$ and dividing by the number of teeth on the planet gear $(\#T)_{planet}$.

$$(TS)_{planet} = (TS)_{in} * (\#T_{ring} / \#T_{planet})$$

Then,

$$GMF = (\#T)_{planet} * (TS)_{planet}$$

Once the gear mesh frequency for the planet gear system is found, any of the turning speeds can be determined by:

Dividing the GMF by the number of teeth on that gear.

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Fractional Gear Mesh

Gear mesh frequency is always present in the vibration signal of a gearbox.

Depending on the wear patterns and the Largest Common Factor (LCF), fractional gear mesh harmonics may appear. If the LCF is 1, only one gear mesh will appear. If the LCF is 2, the ½ gear mesh will appear in the spectrum as the gears become worn. Likewise if the LCF is 3, the 1/3 and 2/3 gear mesh frequency will appear in the spectrum.

Eccentricity also causes fractional harmonics of GMF. If the largest common factor is one, but every other tooth is raised, then again ½ GMF will be present. If a gear has five spokes then it is possible to have five high spots around the gear and 1/5, 2/5, 3/5, 4/5 GMF will appear in addition to the GMF.

Multiples of Gear Mesh

Gear misalignment shows up as harmonics of the gear mesh frequency.

Typically the second and third harmonics are most significant when trending this defect.

If the second gear mesh harmonic is higher in amplitude than the GMF itself, then it is very possible too much backlash exists in the gear set and the gear teeth may be impacting twice during the meshing process: the normal impact during the initial contact and an additional impact during the end of the mesh.

Effect of Load on GMF

The effect of load on the gear set has two contrasting effects on the GMF amplitude depending upon the defect. The general effect of increased load is to increase the amplitude of the gear mesh frequency.

The opposite effect can be expected if the gear has too much backlash present. Too much backlash occurs as the gears become worn and the clearances between the meshing gears increase.

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Sidebands

In gear analysis, sidebands can be valuable in diagnosing gear defects.

Sidebands show up as frequencies on either side of the GMF.

The sideband frequency spacing is equal to the turning speed of either the input shaft speed or the output shaft speed.

The spacing of the sidebands is equal to the turning speed of the gear with the defect. Sidebands appear most commonly because of wear, looseness, and eccentricity.

While the presence of sidebands is important, the amplitude relative to the GMF amplitude is more significant than the amplitude of the GMF.

If the amplitude of the sidebands approaches the amplitude of the GMF, the defect could be severe.

Gear Resonance

One frequency not easy to calculate is the gear resonant frequency. Resonant frequency occurs naturally in all structures, but does not appear in the spectral data unless some other frequency excites the resonance. In gearboxes, eccentricity problems and excessive looseness cause the teeth to mesh with excessive force causing high levels of impacting in the machine that cause the gear resonant frequencies to be excited.

Hunting Tooth Frequency

If a tooth has a manufacturing defect, then it will have a defect frequency associated with it. This Hunting Tooth Frequency (HTF) is subsynchronous as the tooth repetition frequency is less than turning speed. The HTF is simply equal to the product of the GMF and the Largest Common Factor (LCM) between the meshing gears divided by the product of the number of teeth on each gear.

$$HTF = (GMF * LCF) / (\#T_{in} * \#T_{out})$$

This frequency, if present, will be very low and may even be present as a sideband frequency, detectible only by using envelope demodulation. Sometimes HTF is referred to as the tooth repetition frequency.

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Broken Tooth

The effect of a broken tooth is difficult to detect when using only the spectral data.

If one tooth is broken then a pulse will be generated once per revolution of the gear with the broken tooth.

This is simply a 1xTS frequency. The way to detect a broken tooth is to examine the time waveform and look for an impact occurring at a time spacing equal to 1xTS. The time waveform is not sinusoidal but has the impact and ring down once per revolution.

Audible Noise

Unfortunately many gearboxes are considered problems because they are noisy.

However, noise levels are not always a good indication of gear condition.

Very often the GMF or any of the other gear defect frequencies simply excite the natural resonances of the gearbox cover.

This causes the radiated airborne noise levels to increase significantly.

Sometimes lubricating oil gets trapped between the meshing teeth and is forced out at extremely high velocities which can cause the audible noise levels to be high.

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Measurement Considerations

Sensor Selection

Now the spectral data will not do any good if the frequencies of interest are not measured. Several things must be identified before the measurement points are developed.

Which frequencies are important in the analysis of gears?

Low frequencies, such as the hunting tooth frequency all the way to 2x or 3xGMF.

Often it is recommended to set the F_{max} at (2xGMF) + 5xTS in order to see the gear misalignment defect in addition to any sidebands around the 2xGMF.

However, if the maximum frequency selected, F_{max}, is higher than the usable frequency range of the transducer, then a high frequency accelerometer will need to be used in addition to the sensor normally used.

This would mean two measurements taken at the same position. Otherwise, a lower F_{max} could be selected at the cost of unmeasured data. Be sure and use a sensor that accurately measures all the frequencies of interest.

Sensor Attachment

Once the proper sensor has been chosen, make sure the proper attachment method is used. Be aware of the frequency response and the mounting resonances present due to the different mounting methods.

A high-frequency accelerometer attached with a small, but strong magnet may be acceptable. Some cases of very high frequencies, above 10,000 Hz may require stud mounting to get good vibration data.

For spur gears the radial directions provide the most important information because of the direction that the forces are being transmitted though the gears. Helical gears experience a significant amount of axial thrusting and therefore the axial direction contains the best information for the analysis of these gears.

The gearbox covers are not good locations for data collection because of resonances in the gear covers. The bearing locations or the heads of bolts are the most acceptable measurement locations.

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Measurement Point Turning Speed

As the measurement points are being defined for data collection, it is important to realize that the turning speeds will be changing as the speeds are reduced or increased through a gearbox.

If the F_{max} is set to 2xGMF, this could be acceptable for each measurement point along the gear train.

However, pay attention to the lines of resolution that have been selected and adjust them to keep the bandwidth at an acceptable range.

Summary

This section has covered a wide range of topics including the theory of gears, vibration analysis of gears, and measurement point definitions for data collection.

An understanding of the topics covered in this section will lead to a more confident ability to perform vibration analysis of even the most complex gear trains.

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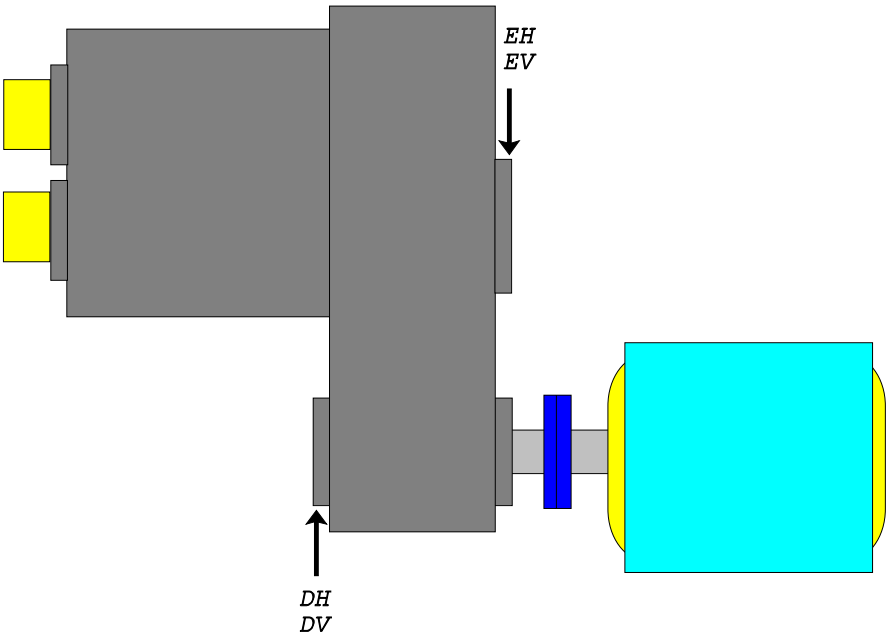
Case Histories

Product Winder Gear Case #1



❶ 1250 HP DC motor driving gearbox. 811 rpm

❷ Input pinion has 24 teeth and meshes with a 72 tooth gear.

❸ Each output shaft has a 24 tooth pinion.

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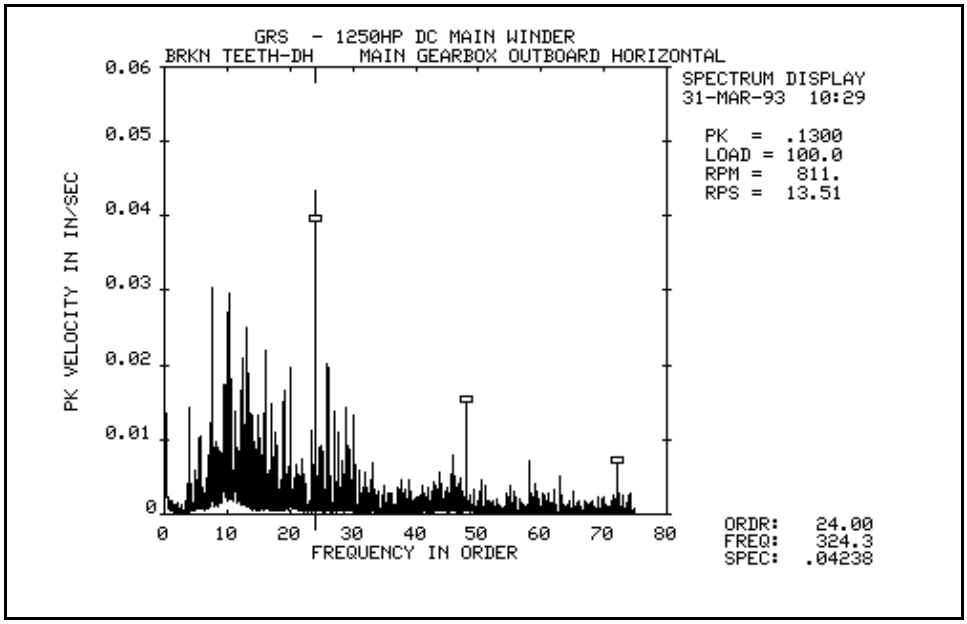
Gear Case #1



The above spectrum was taken on the gearbox at the outboard horizontal position of the input shaft. Shaft speed is 811 rpm.

What is the calculated Gear Mesh Frequency ?

GMF = # T x turning speed

24t x 811cpm = 19,464 cpm (19, 464 divided by 60 = 324.4 Hz)

Gear Mesh Frequency (GMF) is marked by the primary cursor at 24 orders.

1 order = 811cpm (24 orders x 811cpm = 19,464 cpm)

Notice the harmonic cursors are showing the presence of multiples of GMF. The amplitudes of these multiples are low, however, their presence does indicate potential problems.

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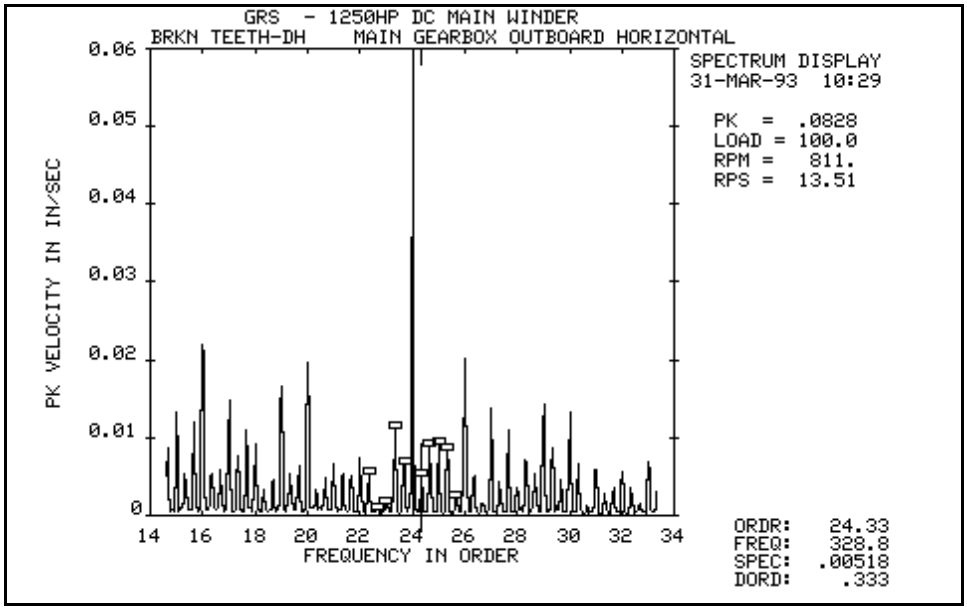
Gear Case #1



This is still looking at the input shaft data.

This spectrum shows the set mark at GMF, with sideband cursors marking peaks at .333 x TS or 1/3 order.

It appears there are many multiples of this sideband. What does this sideband spacing indicate? Recall the case history information stated the reduction ratio in the gearbox is 3:1.

We need to ask ourselves what would create a frequency at 1/3 order of the turning speed of the shaft we are analyzing the data on.

Again the turning speed of this shaft is 811 rpm**.**

811 rpm divided by 3 = 270 rpm

Again, what would create a sideband frequency at 1/3 order (270 rpm) ?

Let’s look at some more data.

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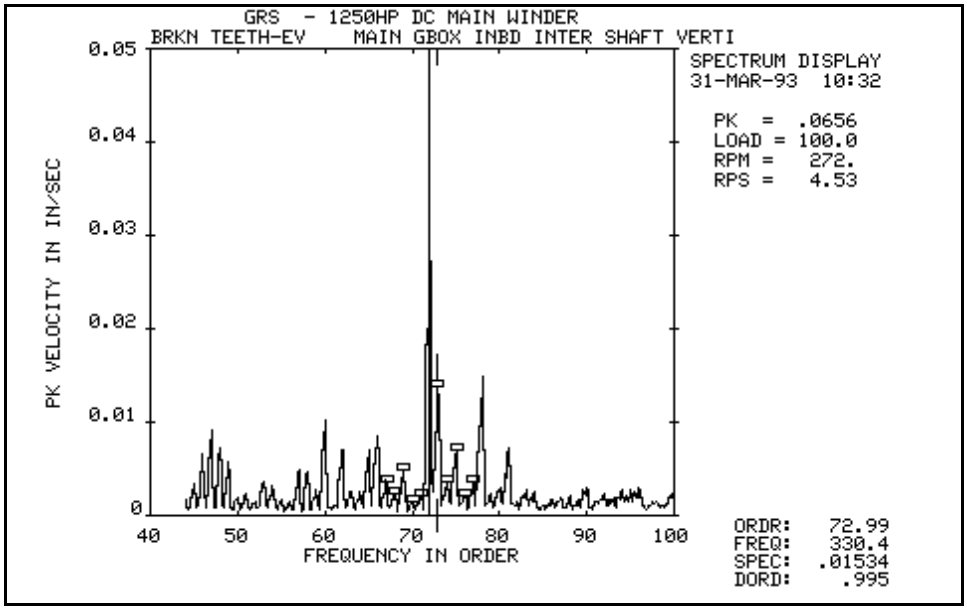
Gear Case #1



This is the intermediate shaft data.

A set mark has been placed on the GMF. The sideband cursor shows sidebands spaced at 1xTS. The spacing of these sidebands determine which shaft has the defective gear.

Check the reference rpm on this measurement point on the spectrum. The reference is listed as 272rpm.

The DORD is listed as .995 orders. Or simply (1 Order).

These sidebands are spaced at the turn speed of this shaft, the data displayed above is of this shaft.

This means that the gear on **"**this shaft**"** has bad teeth on it.

Case solved ?

Not exactly, let's look at some more data

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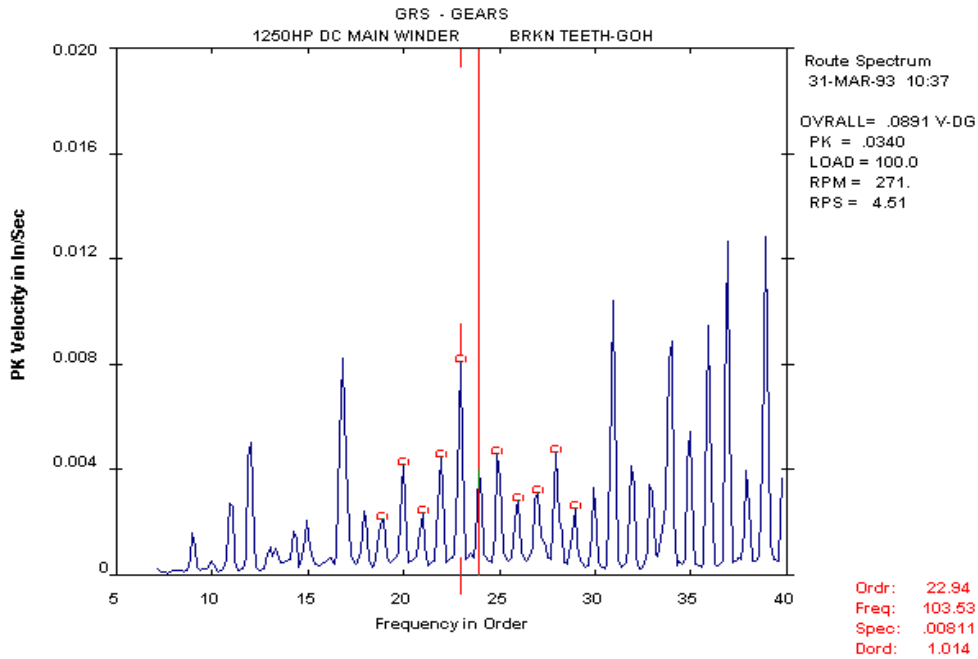
Gear Case #1

Data displayed below is from the last **Output** shaft.



This shaft speed is 270 rpm.

The primary cursor is marking the GMF, the sideband cursors are marking the DORD at 1 order.

This means **this** shaft also has a gear that has bad teeth on it.

When the gear-box was taken down for repairs and inspected for damage, the gear on the intermediate shaft was bad and the gears on the output shaft also were bad.

Now is the Case solved ?

Yes, but we should find the root cause of this failure.

Was it lubrication, poor installation, faulty gears, excessive loads, or did it finally die of old age?

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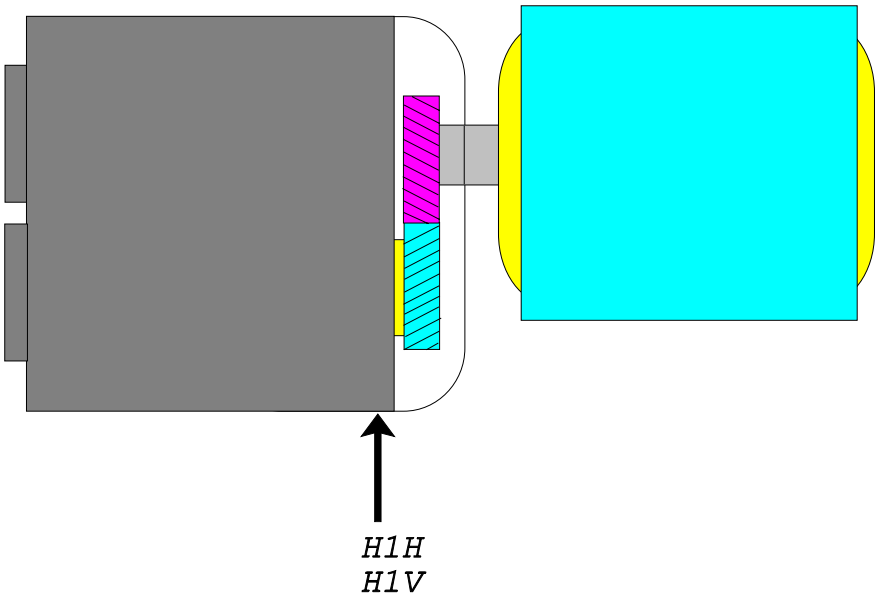
Rotary Screw Compressor Gear Case #2



❶ 500 HP, 1800 RPM Motor.

❷ Compressor is driven by intermittent gearing.

❸ The motor gear has 66 teeth and the compressor gear has 61 teeth.

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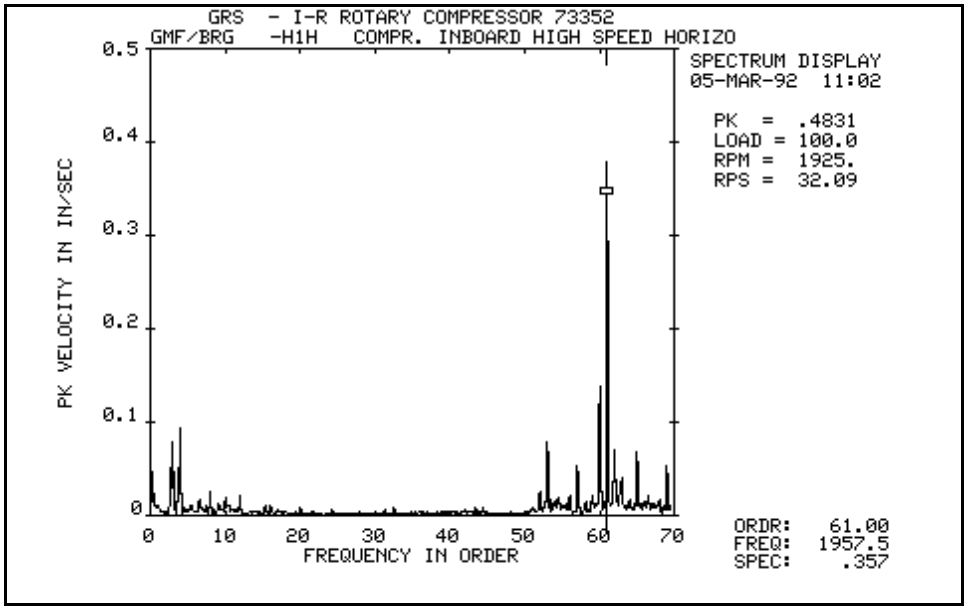
Gear Case #2



The above spectrum shows data collected from the compressor inboard horizontal. A cursor is set on the high speed shaft's GMF (61xTS).

Notice the peaks above and below GMF.

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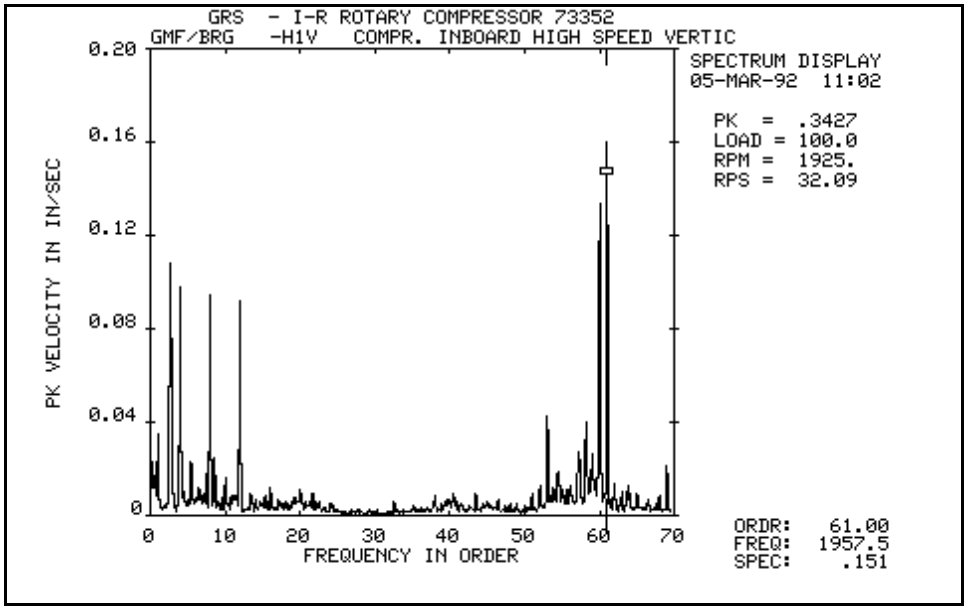
Gear Case #2



The compressor inboard vertical measurement point data is seen above. The cursor has been set on GMF(61xTS). The vertical data also shows the presence of peaks around GMF.

The next page shows the same data with these peaks marked.

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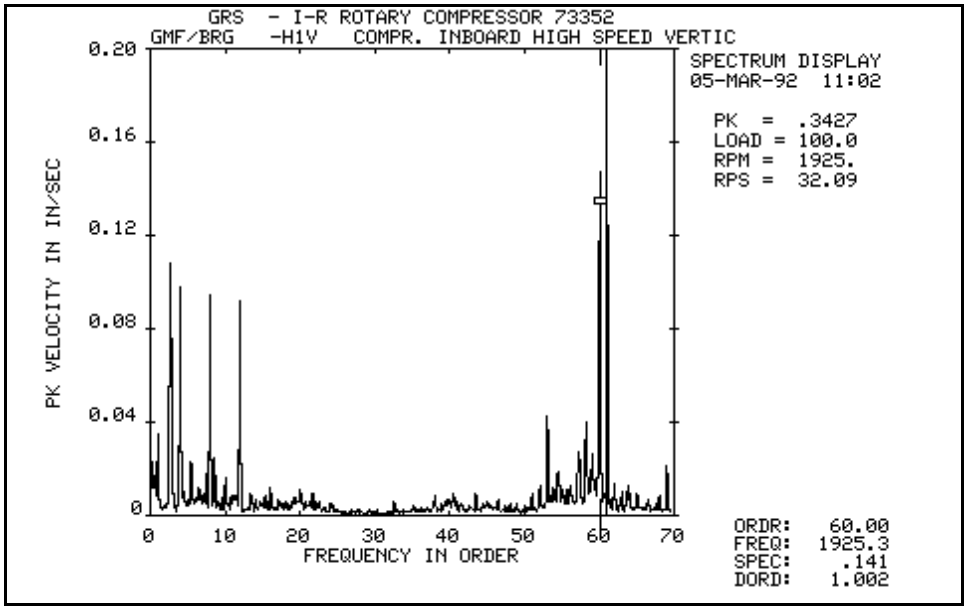
Gear Case #2



A mark has been set on GMF and the cursor marks the highest peak.

The spacing on this peak is equal to 1xTS.

This spacing has determined the defect to be on the compressor gear. Also, notice the amplitude of this sideband compared to the GMF amplitude.

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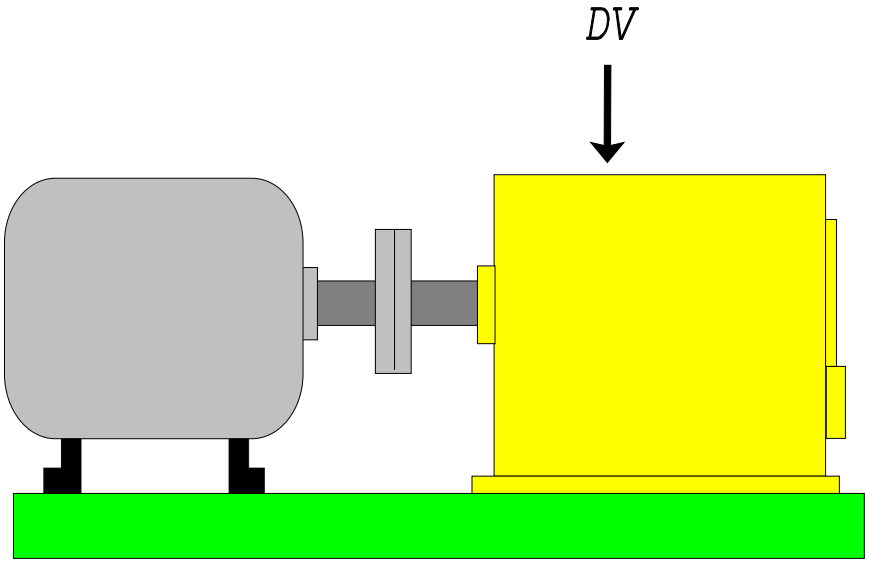
Surge Cake Mixer
Gear Case #3



❶ 75 HP, 1800 RPM motor.

❷ Double-Reduction gearbox direct driven from the motor.

❸ Input pinion has 15 teeth and meshes with a 91 tooth gear.

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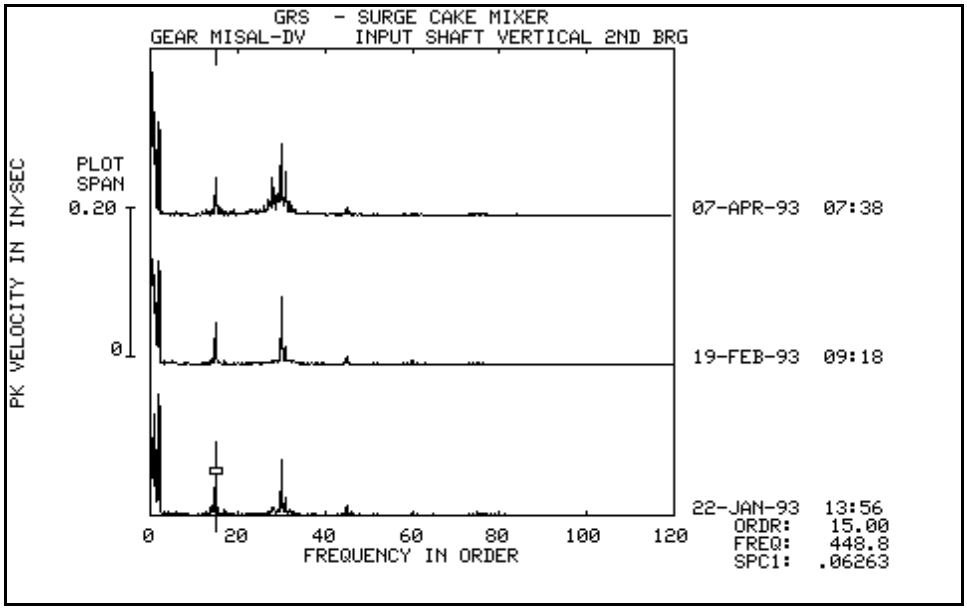
Gear Case #3



The multiple spectrum plot is displayed above from the gearbox outboard vertical point for the input shaft.

The cursor marks GMF (15xTS) of the input shaft.

The data from April shows the peak between 20 - 40 orders has changed.

The next page shows data from February.

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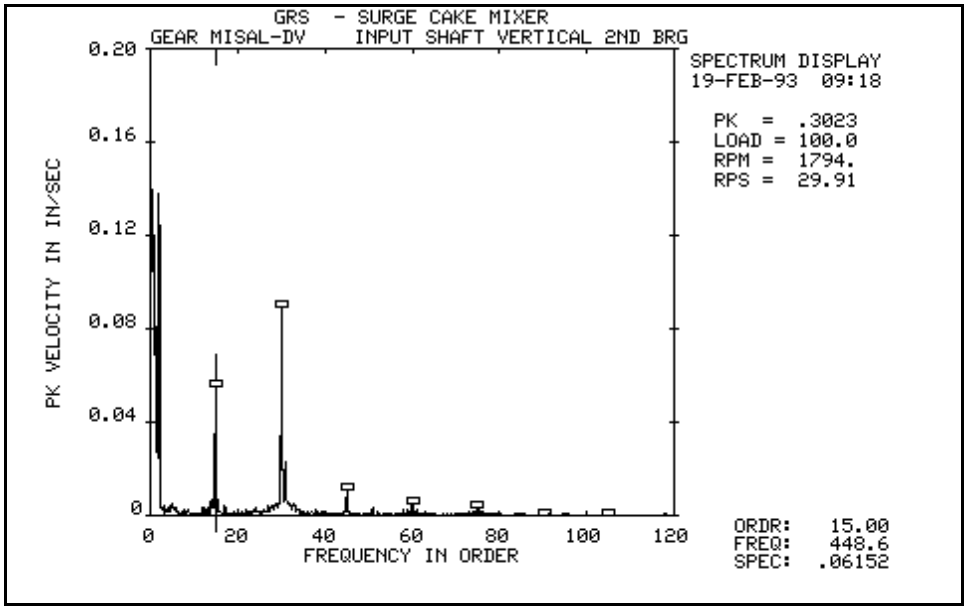
Gear Case #3



A harmonic cursor is set at GMF (15xTS) and shows five multiples of GMF.

Notice the 2x GMF peak is higher than the primary GMF peak.

As with other types of equipment, 2x GMF is an indication of misalignment.

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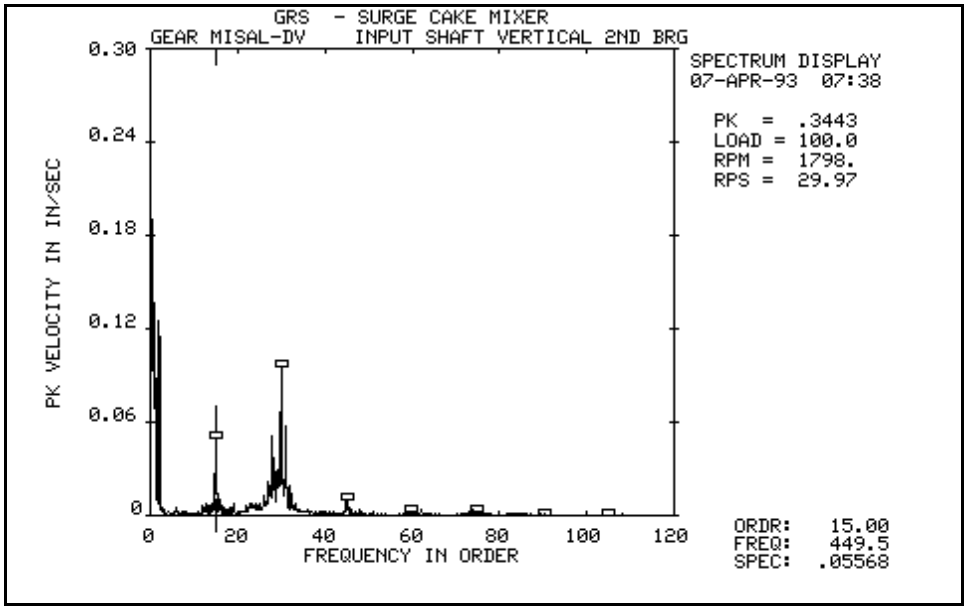
Gear Case #3



The 2x GMF peak has changed from the data collected in February.

It appears the 2x GMF peak has developed sidebands.

The next page shows an expanded view of this group of peaks.

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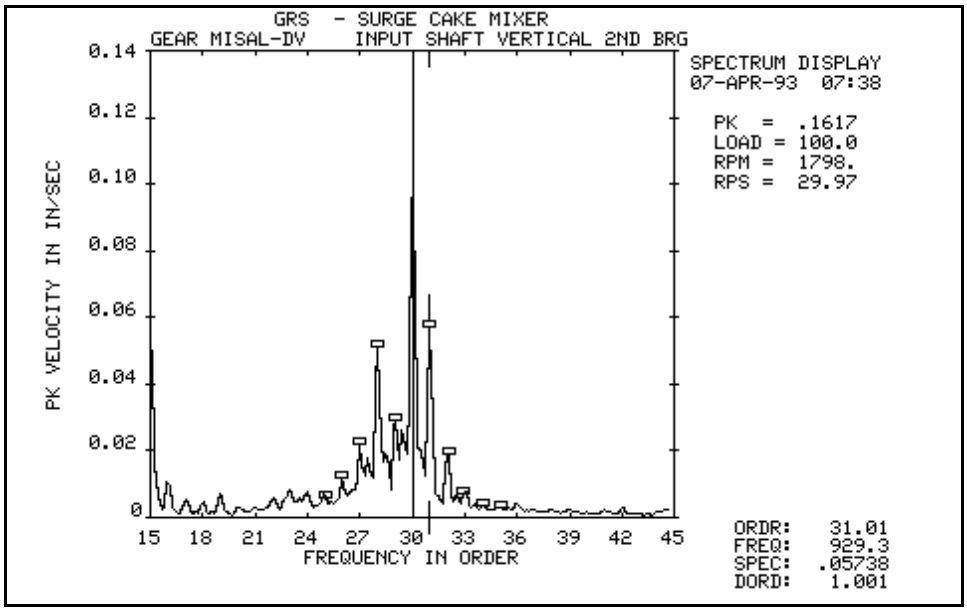
Gear Case #3



A mark has been set at 2x GMF with the sideband cursor showing a spacing of 1xTS.

This spacing indicates a problem with the input shaft pinion.

This unit was sent in for repairs and the input pinion and gear were found to be misaligned.

The misalignment had caused an uneven wear pattern across the face of the gear teeth.

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Section Thirteen
Electrical Defects

Objectives

- Explain the electrical defects associated with AC and DC motors.
- Describe the differences between Induction, Synchronous and DC motors.
- Identify the various electrical defects associated with the above types of motors.
- Explain analysis techniques using single-channel data acquisition.

General Description

Industry utilizes a large number of both AC and DC motors for a wide variety of applications.

These motors can be broken down into two main components:

The **rotor** and the **stator**.

How these two components electrically interact will determine the characteristics associated with specific electric motor defects.

Although most vibration problems associated with electrical systems are motor related, do not overlook other sources.

- Loose laminations in the power transformer
- SCR pulses in the speed controller
- Unbalanced supply voltage
- High currents from welders
- Solenoids

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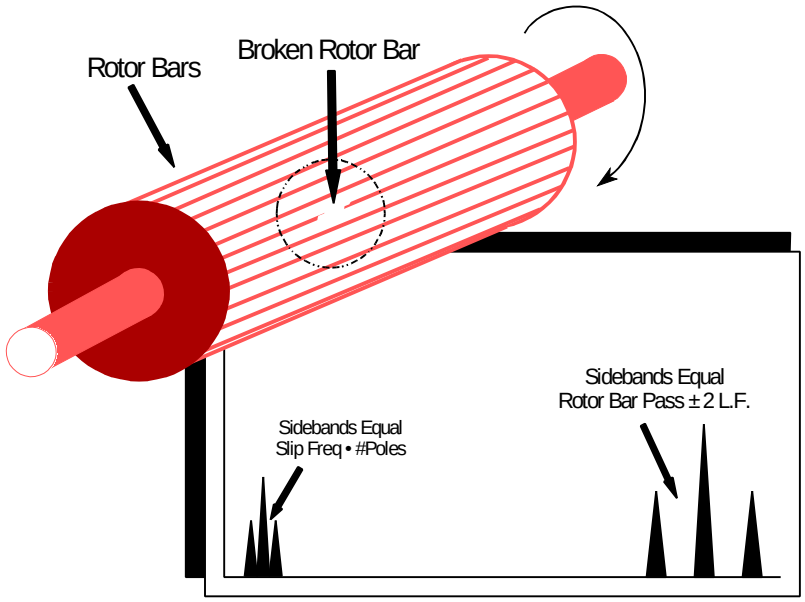
AC Induction Motor Analysis

Rotor Related Defects

Broken or Open Rotor Bar



Broken or Open Rotor Bars

The predominant vibration appears at 1xTS with sidebands spaced at a frequency equal to the number of poles on the motor multiplied by its slip frequency.

Rotor slot frequency may also be present with sidebands spaced at 2x line frequency.

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For the Vibration Analyst to be proficient at calling electrical problems on their equipment it is essential the calculations below are understood.

Please consider the following:

Slip Frequency = Magnetic Field Freq. - Rotor Freq.

The magnetic field frequency is the rated speed listed on the motor tag.

The rotor frequency is the actual speed of the motor.

Example:
Rated motor speed = 1800 rpm
Actual motor speed = 1785 rpm

Slip Frequency = 1800 - 1785 = 15 rpm

Rotor Slot Frequency = # of Rotor Slots * Rotor Freq.

The # of Slots = how many slots the rotor has on it

Rotor Frequency = the actual speed of the rotor

Example:
44 slots on rotor x 1785 = 78,540 cpm = Rotor Slot Frequency

Stator Slot Frequency = # of Stator Slots * Rotor Freq.

The # of slots on the Stator x actual speed on the rotor

Example:
46 slots on the Stator x 1785 = 82,110 cpm = Stator Slot Frequency

---

# of Motor Poles = [Line Freq(Hz) * 120] / Magnetic Field Freq(CPM)

Example:
60 Hz x 120 = 7200 divided by 1800 = 4 poles

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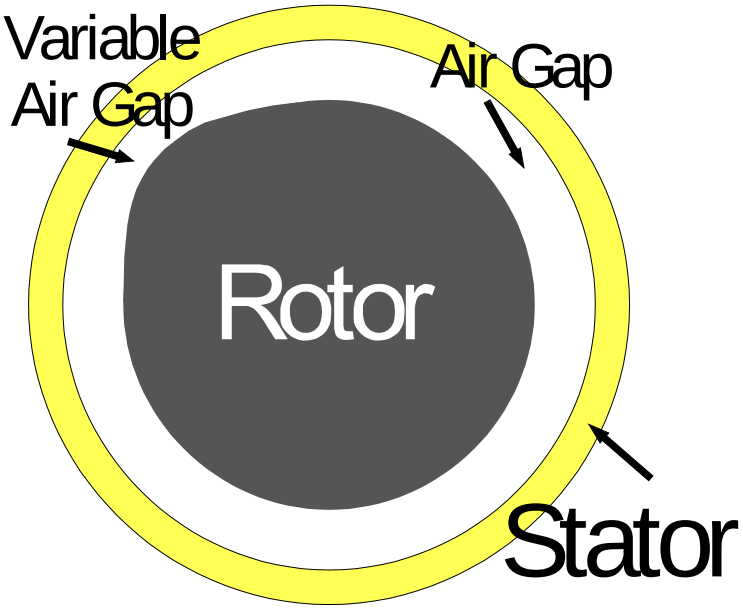
The waveform resembles that for imbalance, because the rotor actually is out of balance. In a live, real time, display of the waveform, however, you can see the electrically induced vibration disappear immediately when you turn off the power to the motor. The signal disappearance helps you differentiate this fault from mechanical imbalance, where the vibration would coast down with shaft speed.

Loose Iron or Slot

The primary vibration appears in the spectrum at 2xLF and slot frequency with sidebands spaced at 2xLF. The waveform shows a dominant cycle of 2xLF.



Dynamic Eccentricity

The principal vibration appears in the spectrum at 1xTS with sidebands spaced at slip frequency multiplied by the number of motor poles and/or 2xLF.

Slot frequency may also be present with sidebands spaced at 2xLF. The waveform resembles that for imbalance, because the rotor is actually out of balance. In a live, real time display of the waveform, however, you can see the electrically induced vibration disappear immediately when you turn off the motor power. The signal disappearance (when power is shut off) helps you distinguish this fault from mechanical imbalance.

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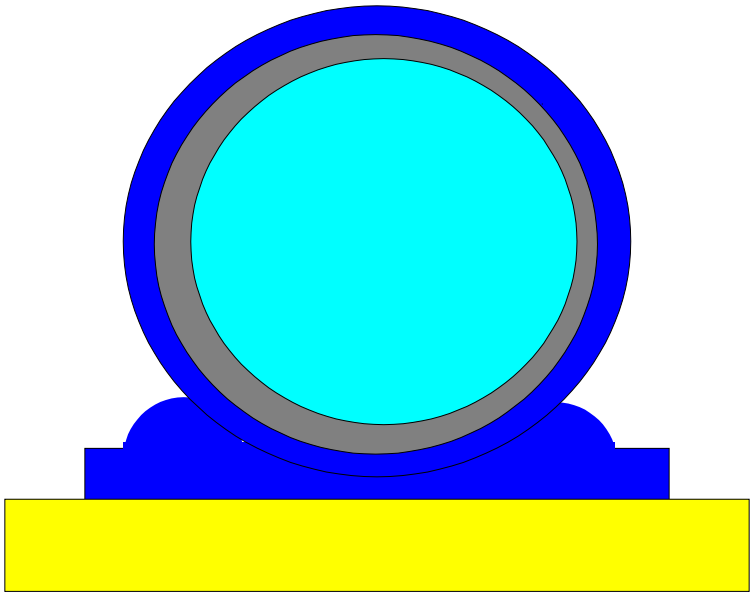
Stator Related Defects



Static Eccentricity

Static eccentricity is also known as **stator eccentricity**.

An eccentric stator will cause an unequal air gap between the rotor and the stator. This results in the spectrum having a primary peak at 2xLF that is very directional. Slot frequency with sidebands spaced at 2xLF may also be present. A poorly built stator, improper lamination, stamping, loose laminations, warped motor base or a soft foot are all possible causes of this defect. The waveform demonstrates a dominant cycle at a frequency related to the 2xLF. As with other electrical faults, you can confirm this defect by monitoring the signal in a live, real time waveform and securing the motor. The energy should disappear immediately.

If a soft foot is suspected, monitor 2xLF in a live spectrum and loosen one motor foot bolt at a time observing any change in the amplitude at 2xLF. If the amplitude at 2xLF is reduced during this procedure, then a motor soft foot is probable. This test does not determine which of the motor feet has the potential problem.

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Loose Stator Laminations or Loose Stator Core

The primary vibration appears in the spectrum at 2xLF and harmonics of 2xLF.

The waveform demonstrates cycles at frequencies related to 2xLF. As with other electrical faults, you can confirm this problem by monitoring the signal in a live or real time waveform and turning off the power. The energy should disappear immediately.

Open or Shorted Windings

The primary vibration appears in the spectrum at 2xLF and increases in amplitude as motor temperature climbs.

A dominant cycle at a frequency associated with 2xLF is shown in the waveform**.**

This problem can be confirmed by monitoring the live waveform and securing the power to the motor. The energy should disappear immediately.

Insulation Breakdown and Unbalanced Phase

Both appear in the spectrum at 2xLF. The waveform demonstrates a dominant cycle at a frequency related to the 2xLF.

This fault can be verified by monitoring the real time waveform and turning off the power to the motor. The energy should disappear immediately.

Loose Connector

The primary frequency appears in the spectrum at 2xLF with sidebands spaced at one-third line frequency.

The waveform shows a dominant cycle at a frequency identified with 2xLF. This problem can be particularly difficult to identify if the loose connector is intermittently making contact.

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AC Synchronous Motor Analysis

A synchronous motor will have a running speed of exactly the magnetic field frequency of the stator.

This is accomplished by designing the rotor as a fixed pole magnet.

This rotating magnet can be a permanent magnet but is usually an electromagnet energized from an external direct current source through slip rings.

The rotating magnetic field on the stator and the fixed magnetic field on the rotor lock in and cause the rotor to turn at synchronous speed.

Rotor Related Defects

Unequal Pole Resistance

The primary vibration appears in the spectrum at 2xTS. The waveform demonstrates a dominant cycle at frequency related to 2xTS.

This fault can be verified by monitoring the real time waveform and securing the rotor field. The energy should disappear immediately.

Dynamic Eccentricity

The principal vibration appears in the spectrum at 1xTS.

A 2xLF peak or stator slot frequency with sidebands spaced at 2xLF may also be present.

The waveform resembles that for imbalance, because the rotor is actually out of balance.

In a live, real time display of the waveform, however, you can see the electrically induced vibration disappear immediately when power is secured to the motor. The signal disappearance helps you distinguish this fault from mechanical imbalance.

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Stator Related Defects

Loose Stator Laminations or Loose Stator Core

The primary vibration appears in the spectrum at 2xLF and harmonics of 2xLF.

The waveform demonstrates cycles at frequencies related to 2xLF. As with other electrical faults, you can confirm this problem by monitoring the signal in a live, or real time waveform and turning off the power. The energy should disappear immediately.

Open or Shorted Windings

The primary vibration shows up in the spectrum at 2xLF and increases in amplitude as motor temperature climbs.

A dominant cycle at a frequency associated with 2xLF is shown in the waveform**.** This problem can be confirmed by monitoring the live waveform and shut off the power to the motor. The energy should disappear immediately.

Insulation Breakdown and Unbalanced Phase

Both appear in the spectrum at 2xLF.

The waveform demonstrates a dominant cycle at a frequency related to the 2xLF. This fault can be verified by monitoring the real time waveform and turning off the power to the motor. The energy should disappear immediately.

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Static Eccentricity

Static eccentricity is also known as stator eccentricity.

An eccentric stator will cause an unequal air gap between the rotor and the stator. This results in the spectrum having a primary peak at 2xLF that is very directional.

Slot frequency with sidebands spaced at 2xLF may also be present.

A poorly built stator, improper lamination, stamping, loose laminations, warped motor base, or a soft foot are all possible causes of this defect.

The waveform demonstrates a dominant cycle at a frequency related to the 2xLF. As with other electrical faults, you can confirm this defect by monitoring the signal in a live, real time waveform and securing the motor.

The energy should disappear immediately.

Loose Connector

The primary vibration appears in the spectrum at 2xLF with sidebands spaced at one-third line frequency.

The waveform shows a dominant cycle at a frequency identified with 2xLF.

This problem can be particularly difficult to identify if the loose connector is intermittently making contact.

Variable Frequency Drives For AC Induction Motors

The need to vary machine operating speeds is required in certain industrial applications.

Variable Frequency Drives (VFDs) or Inverters are commonly used to control the speed of an AC induction motor.

The operating speed of the motor is solely dependent on the output from the VFD. The motor speed will vary directly as the output from the VFD increases or decreases.

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Rotor Related Defects

Broken or Open Rotor Bars

The predominant vibration appears at 1xTS with sidebands spaced at a frequency equal to the number of poles on the motor multiplied by its slip frequency.

Rotor slot frequency may also be present with sidebands spaced at 2x line frequency. Please consider the following:

Slip Frequency = Magnetic Field Freq. - Rotor Freq.

Rotor Slot Frequency = # of Rotor Slots * Rotor Freq.

Stator Slot Frequency = # of Stator Slots * Rotor Freq.

# of Motor Poles = (Line Freq. * 120) / Magnetic Field Freq.

The waveform resembles that for imbalance, because the rotor actually is out of balance.

In **Monitor** mode you can see the electrically induced vibration disappear immediately when you turn off the motor power. The signal disappearance helps you differentiate this fault from mechanical imbalance, where the vibration would coast down with shaft speed.

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Loose Iron or Slot

The primary vibration appears in the spectrum at 2xLF and slot frequency with sidebands spaced at 2xLF. The waveform shows a dominant cycle of 2xLF.

Dynamic Eccentricity

The principal vibration appears in the spectrum at 1xTS with sidebands spaced at slip frequency multiplied by the number of motor poles and/or and 2xLF.

Slot frequency may also be present with sidebands spaced at 2xLF.

The waveform resembles that for imbalance, because the rotor is actually out of balance. In a live, real time display of the waveform, however, you can see the electrically induced vibration disappear immediately when you turn off the motor power. The signal disappearance helps you distinguish this fault from mechanical imbalance.

Stator Related Defects

Loose Stator Laminations or Loose Stator Core

The primary vibration appears in the spectrum at 2xLF and harmonics of 2xLF.

The waveform demonstrates cycles at frequencies related to 2xLF. As with other electrical faults, you can confirm this problem by monitoring the signal in a live or real time waveform and turning off the power. The energy should disappear immediately.

Open or Shorted Windings

The primary vibration shows up in the spectrum at 2xLF and increases in amplitude as motor temperature climbs.

A dominant cycle at a frequency associated with 2xLF is shown in the waveform**.** This problem can be confirmed by monitoring the live waveform and securing the power to the motor. The energy should disappear immediately.

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Insulation Breakdown and Unbalanced Phase

Both appear in the spectrum at 2xLF.

The waveform demonstrates a dominant cycle at a frequency related to the 2xLF. This fault can be verified by monitoring the real time waveform and turning off the power to the motor. The energy should disappear immediately.

Static Eccentricity

Static eccentricity is also known as **stator eccentricity**.

An eccentric stator has an unequal air gap between the rotor and the stator.

This results a very directional spectrum with a primary peak at 2xLF. Slot frequency with sidebands spaced at 2xLF may also be present.

A poorly built stator, improper lamination, loose laminations, stamping, warped motor base or a soft foot are all possible causes of this defect.

The waveform demonstrates a dominant cycle at a frequency related to the 2xLF. As with other electrical faults, this defect can be confirmed by monitoring the signal in a live, real time waveform and securing the motor. The energy should disappear immediately.

Loose Connector

The primary frequency appears in the spectrum at 2xLF with sidebands spaced at one-third line frequency.

The waveform shows a dominant cycle at a frequency identified with 2xLF. This problem can be particularly difficult to identify if the loose connector is intermittently making contact.

Variable Frequency Drive Defects

Open or Shorted SCR

The primary vibration appears in the spectrum at multiples of line frequency. A dominant cycle at a frequency associated with line frequency is shown in the waveform. Further electrical testing is recommended if this defect is suspected. This could include jumpering out or replacing one set of SCRs at a time.

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DC Motor Analysis

A DC motor consists of a stationary component called the ***field***.

The field windings create the magnetic North and South poles on the stator.

The rotating component of the DC motor is called the ***armature***. The armature winding current is supplied through brushes and creates the magnetic field on the rotor.

A commutator is attached to the ends of the armature coil and is used to reverse the current flow in the coil.

All DC motors operate on the principle of **Motor Action** which states: a current carrying conductor placed in a magnetic field tends to move at right angles to the field either up or down.

DC motors are classified by the way the main field is connected to the armature. This will determine how the motor reacts to different types of faults.

Electrically testing the motor will identify most of the electrical faults associated with the field and armature.

Rotor Related Defects

Commutator or Brush Problems

The primary vibration tends to show up in the spectrum at the (number of commutator segments) x (number of brush rows) x (shaft turning speed). A dominant frequency associated with the commutator or brush defect is shown in the waveform.

DC Drive Defects

Open or Shorted SCR

The primary vibration appears in the spectrum at multiples of line frequency. A dominant cycle at a frequency associated with line frequency is shown in the waveform. Further electrical testing is recommended if this defect is suspected. This could include jumpering out or replacing one set of SCRs at a time.

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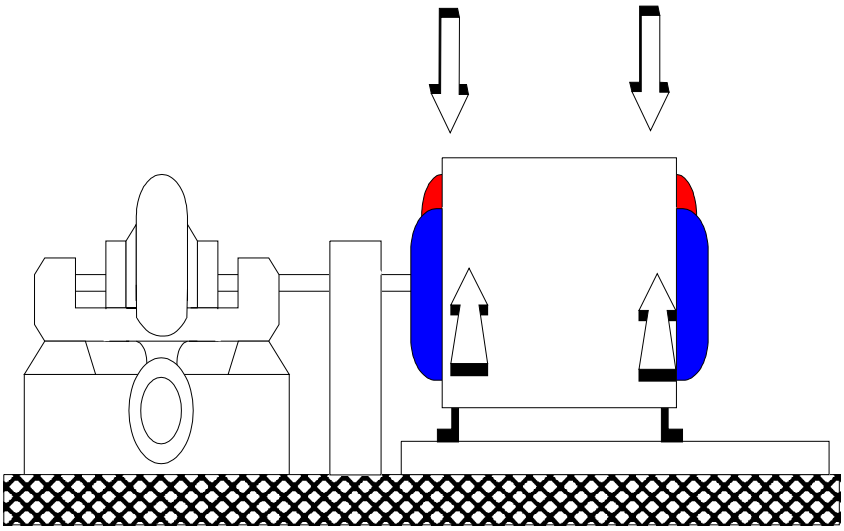
Case Histories

Boiler Feed Pump
Electrical Defect Case #1



❶ 300 HP, 3600 RPM AC Induction Motor.

❷ The motor supply breaker was intermittently tripping.

❸ Speed fluctuations of the motor were observed.

❹ 47 amp surges were measured with clamp on ammeter.

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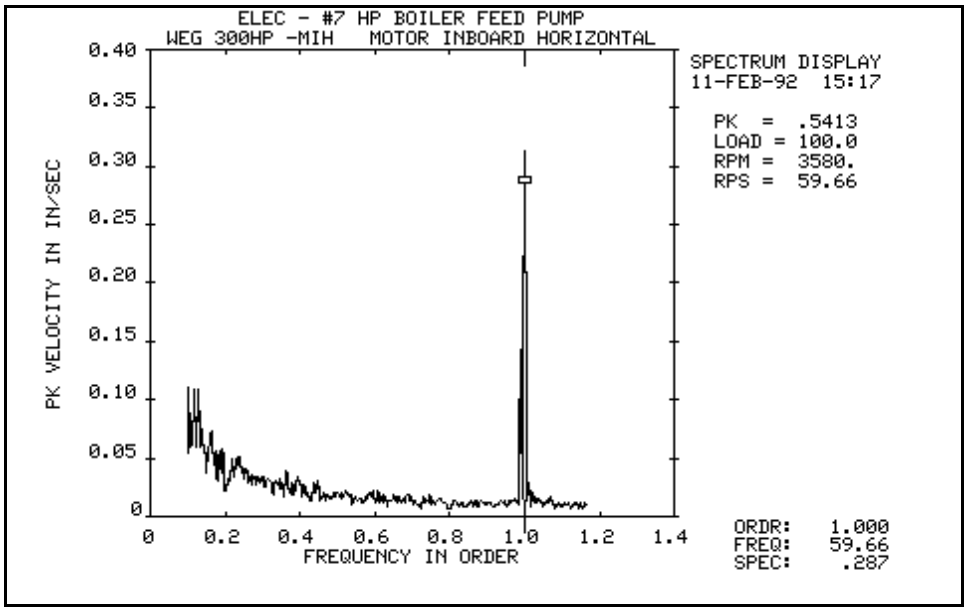
Boiler Feed Pump
Electrical Defect Case #1



The spectrum above comes from a boiler feed pump that was suspected of having an electrical fault. The vibration data was collected after the stator was tested and determined to be in satisfactory condition.

The cursor is marking the running speed peak of the motor. Notice the skirt width of the running speed peak.

Also note that we are only looking out to 1.4 orders.

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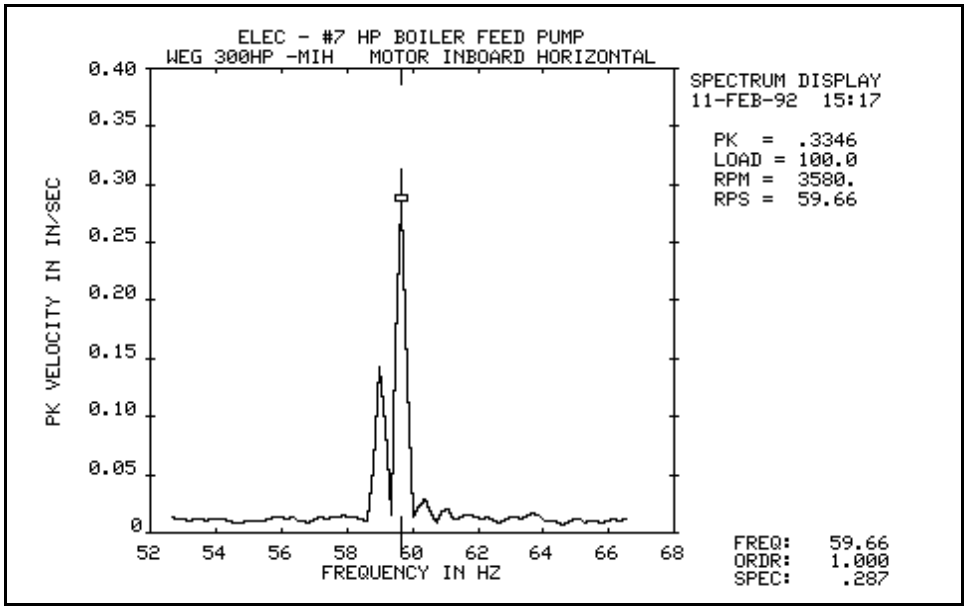
Boiler Feed Pump
Electrical Defect Case #1



When the spectrum from the previous page is expanded, the sideband around motor turning speed is now more obvious.

This may have been identified as unbalance if a high resolution spectrum was not taken. This demonstrates the importance of collecting high resolution data when an electrical defect is suspected.

What could be the possible cause of the motor running speed sideband?

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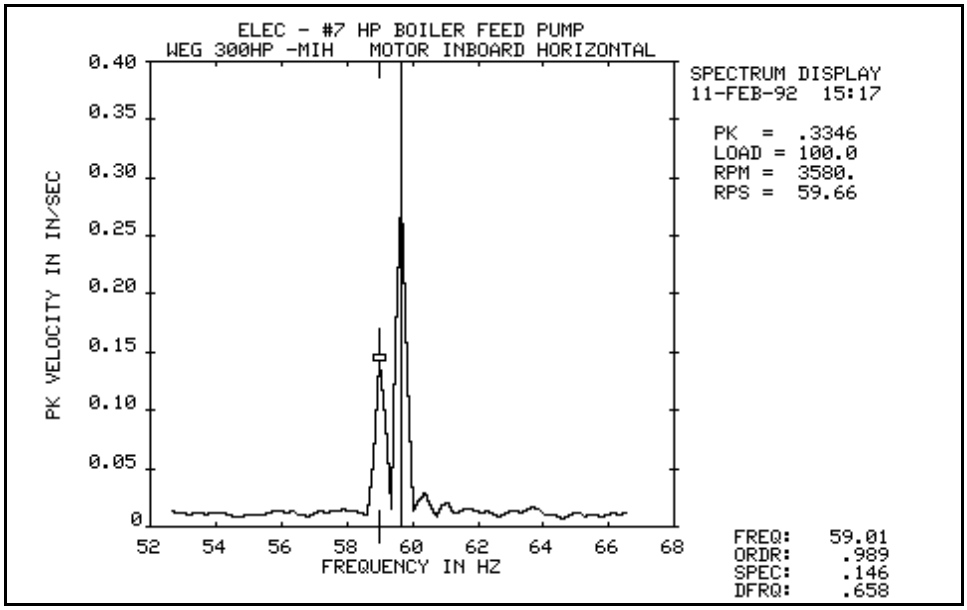
Boiler Feed Pump
Electrical Defect Case #1



A mark has been set on the motor 1xTS peak. The difference frequency of the sideband is 0.66 Hz, which equals the number of motor poles (2) times the slip frequency (0.34 Hz).

Slip Frequency	=	Magnetic Field Speed - Run Speed
	=	60 Hz - 59.66 Hz
	=	0.34 Hz

Sideband	=	Number of Motor Poles x Slip Frequency
	=	2 X 0.34 Hz
	=	0.68 Hz

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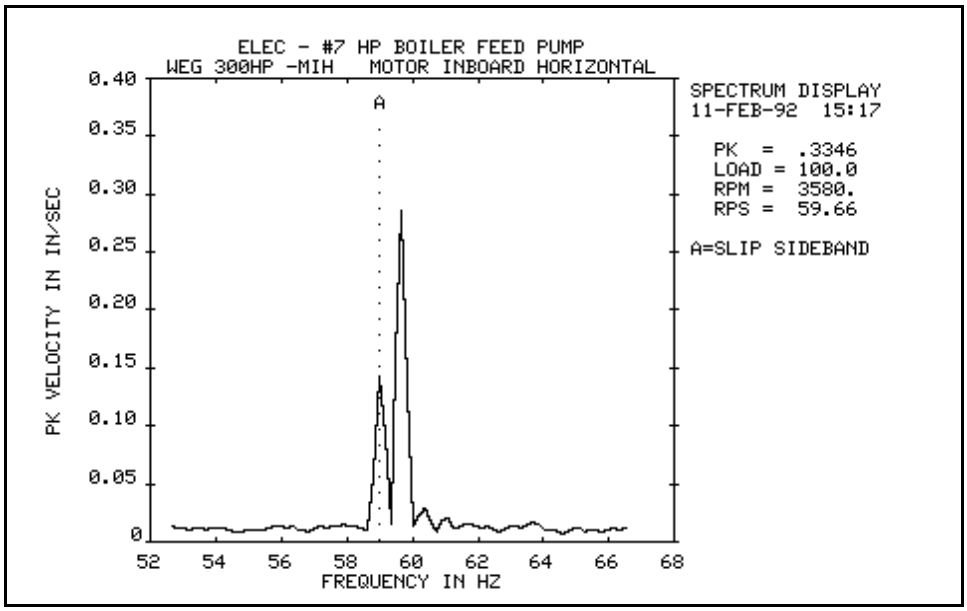
Boiler Feed Pump
Electrical Defect Case #1



The sideband spacing of 0.66 Hz around 1xTS of the motor approximates this frequency.

The presence of this sideband is a good indication of a rotor bar defect.

A current measurement analysis would confirm if a rotor bar problem exists. In this case, **the repair shop found five open rotor bars on this motor**.

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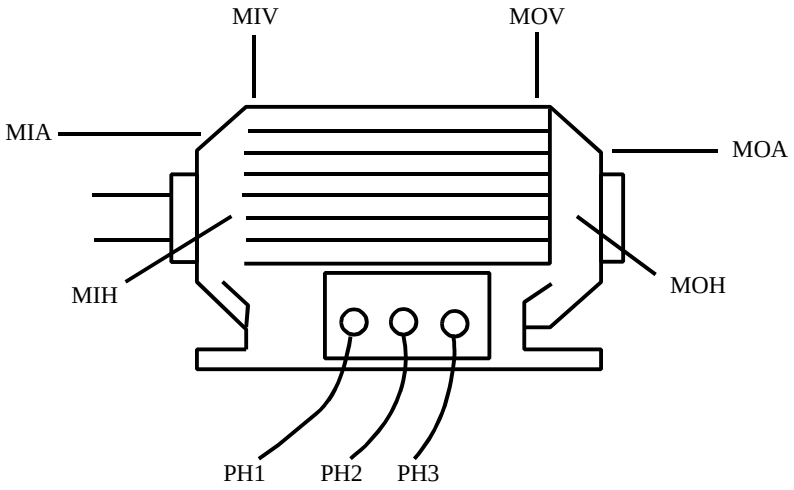
Case #2 Inverter Driven Induction Motor



❶ This 50-HP, four-pole motor is powered by an inverter, so the line frequency is not necessarily at 60 Hz.

❷ The motor drives an exhaust fan in a painting booth.

❸ In addition to vibration readings, current measurements were also collected on Phase 1 of the motor.

❹ This is a new motor. Other motors supplied at the same time did not have these same vibration characteristics.

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The spectrum below comes from the inverter driven motor.

Examine the spectrum to find the 1xTS peak at 26.1 Hz or 1566 RPM.

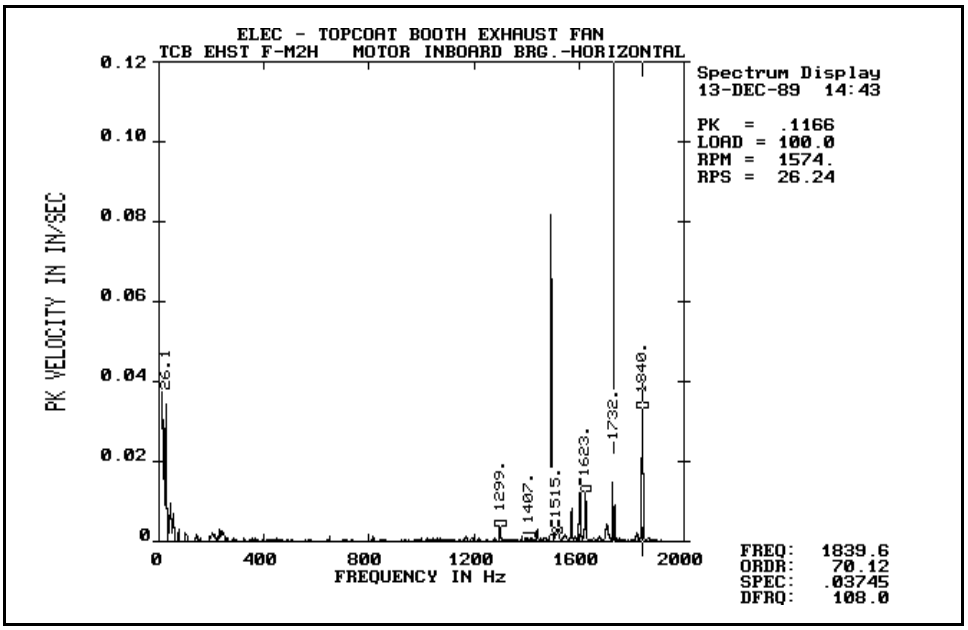
Note the high frequency energy around 1700 Hz. This energy first seems to indicate a bearing fault.

Further examination shows a peak at 1732 Hz, which is 66 orders of TS. This value is probably the **rotor bar pass frequency**, which means the rotor has 66 rotor bars.

The **line frequency is 54.1 Hz, so 2xLF equals 108.2 Hz.**

This spectrum shows very clear sidebands of 108.0 Hz around the peak at 1732 Hz.

Therefore, these peaks are sidebands of 2xLF around the **rotor bar pass frequency**. These peaks indicate an electrical problem.



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The line frequency appears at 54.1 Hz.

The current amplitude at turning speed approximates 41 dB amperes.

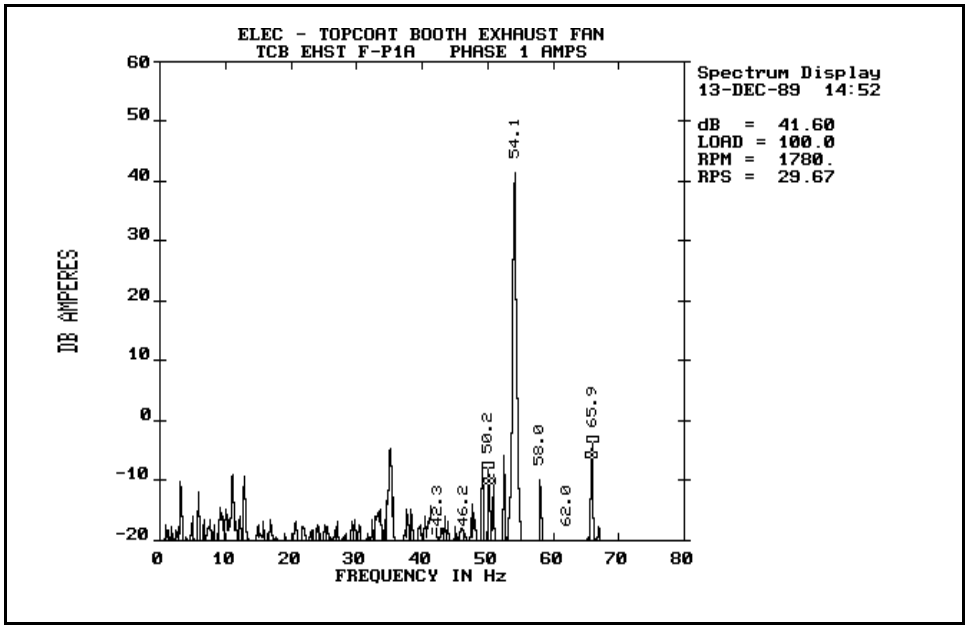
From the previous page, the motor turning speed is 26.1 Hz.

Because this is a four-pole motor, the magnetic field speed or synchronous speed would be half the line frequency or 27.05 Hz.

$$\textit{Slip frequency} = \textit{Magnetic field speed} - \textit{run speed}$$

$$\textit{Slip frequency} = 27.05 \text{ Hz} - 26.10 \text{ Hz}$$

$$\textit{Slip frequency} = 0.95 \text{ Hz} \quad (57 \text{ RPM})$$



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The sidebands in the current spectrum should occur at a spacing equal to the number of motor poles times the slip frequency.

$$\textit{Sideband} = \textit{Number of motor poles} \times \textit{slip frequency}$$

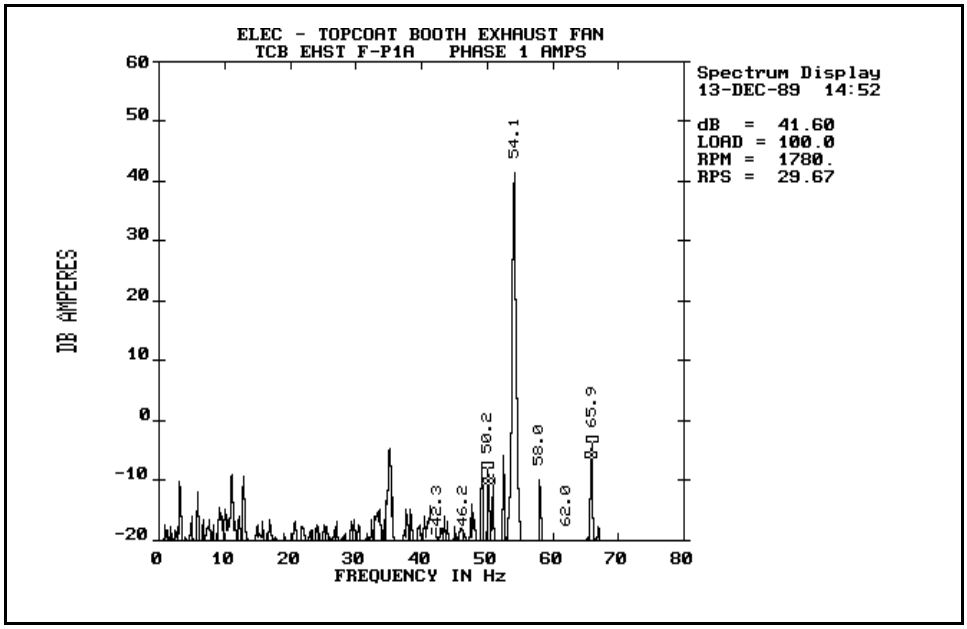
$$\textit{Sideband} = 4 \times 0.95\ \textit{Hz}$$

$$\textit{Sideband} = 3.8\ \textit{Hz}$$

The sideband spacing of 3.9 Hz nearly equals this frequency.

The reason for examining the current spectrum in dB amperes is to see these sidebands.

If the sidebands are less than 50 dB down from the line frequency peak, then suspect a rotor bar problem. In this case, the sidebands at 50.2 Hz and 65.9 Hz are less than 50 dB down.



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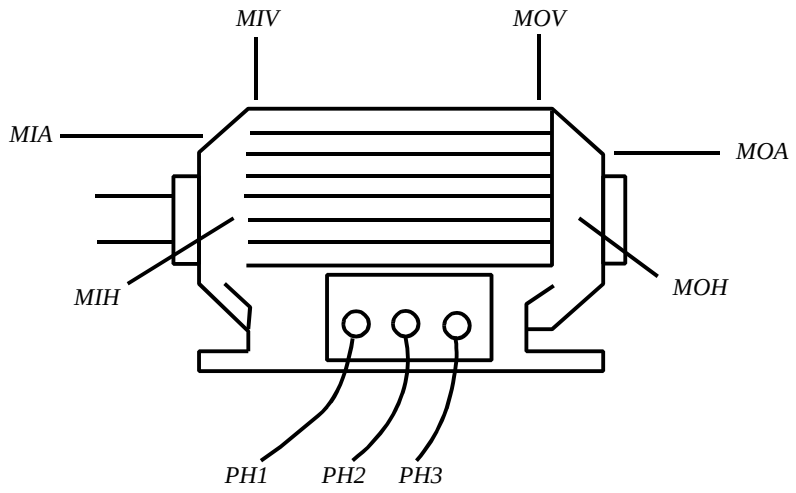
Two-Pole Induction Motor



❶ This 50 HP, two pole induction driven motor operates on 60 Hz line frequency power.

❷ The actual running speed of this motor is 59.54 Hz, or 3572.4 RPM.

❸ This example demonstrates the need for high resolution spectra when troubleshooting electrical faults.

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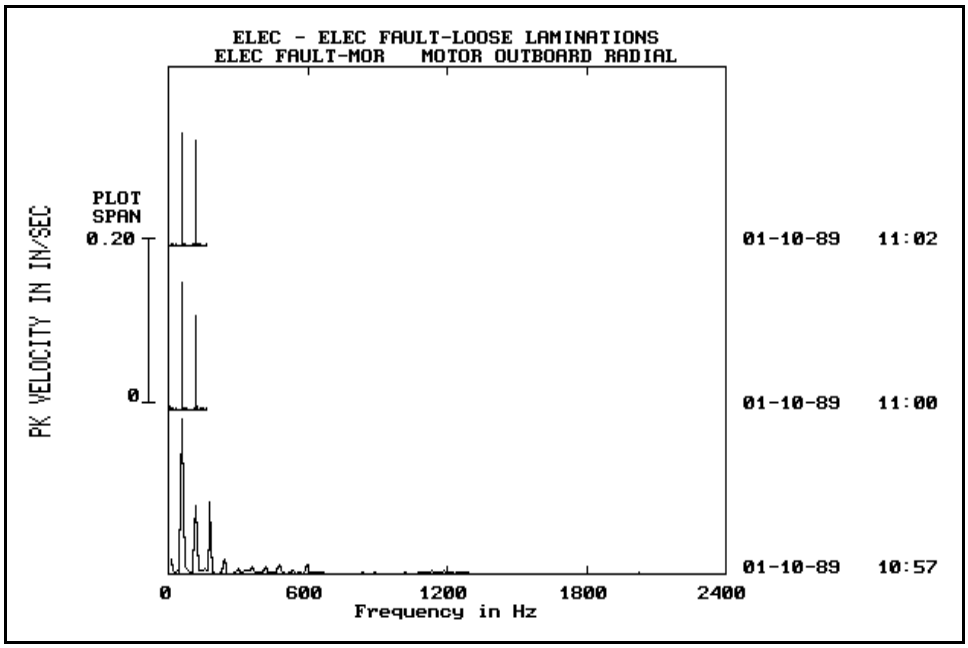
You need high resolution spectra to diagnose electrical faults.

The 400-line FFT below comes from the Route mode of the Model 2110/2115 Machinery Analyzer.

The 800 and 1600 line FFTs come from Acquire Spectrum in the Analyze mode of the Model 2110/2115.



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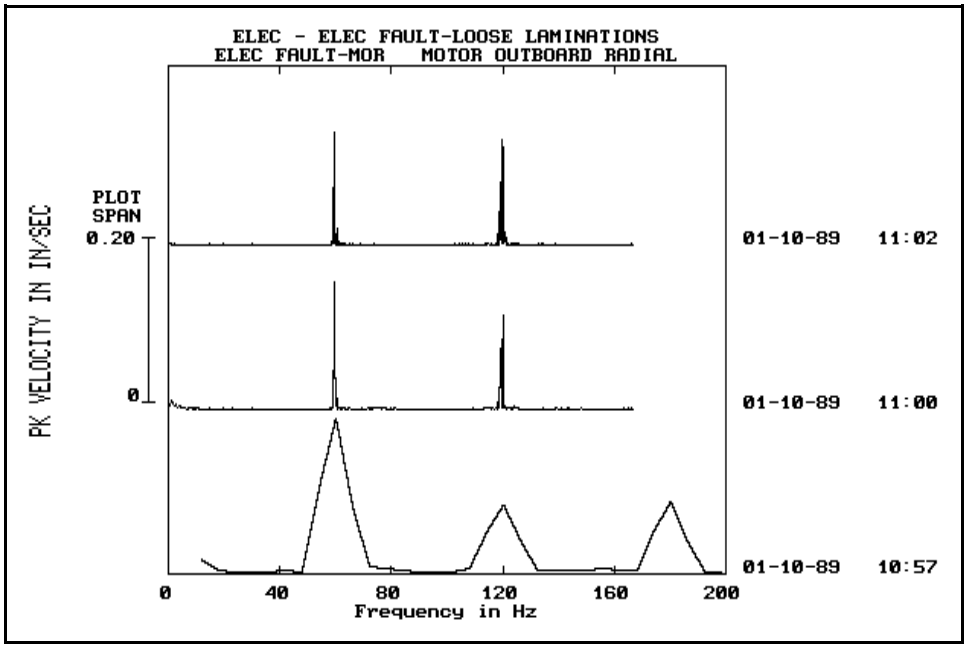

The bottom spectrum in the multiple-spectra plot above shows only broad peaks at 1xTS and 2xTS.

The top two spectra indicate that many closely spaced peaks combine to form those broad peaks in the bottom spectrum.

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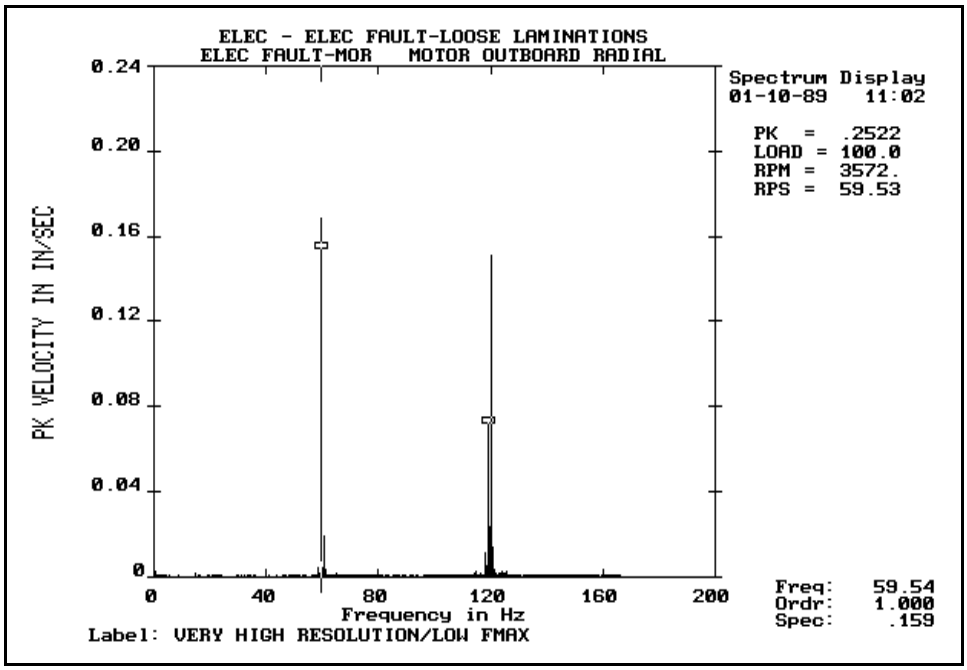

The spectrum above shows a 1600-line FFT.

Note that the 2xTS harmonic cursor is not on the highest peak.

Another peak, located at a frequency just above the 2xTS frequency, has a much higher amplitude.

The plot on the next page expands the frequency scale of the spectrum above. The expansion locates 120 Hz at the center of the horizontal (frequency) scale.

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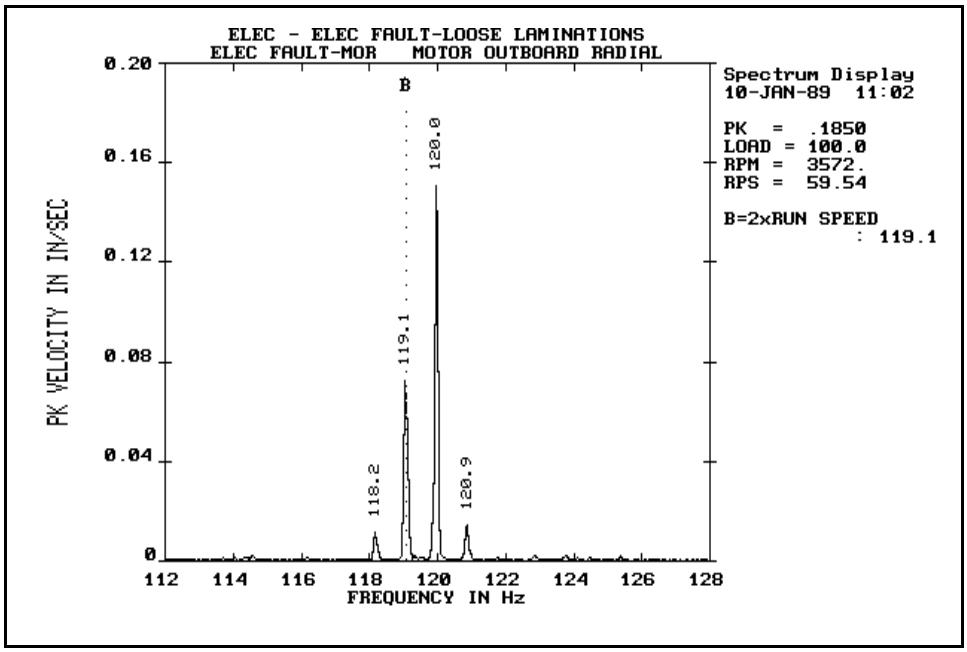

When you expand the plot on the previous page, the peak at 120 Hz shows up more clearly.

This major peak is located at exactly 120 Hz or 2xLF. Note that the 2xTS peak of the motor actually occurs at 119.1 Hz.

The sidebands are spaced at about 0.9 Hz, which equals the number of poles on the motor (2) times the slip frequency (0.46 Hz).

slip frequency	=magnetic field speed - run speed
=	60 Hz - 59.54 Hz
=	0.46 Hz (27.6 RPM)

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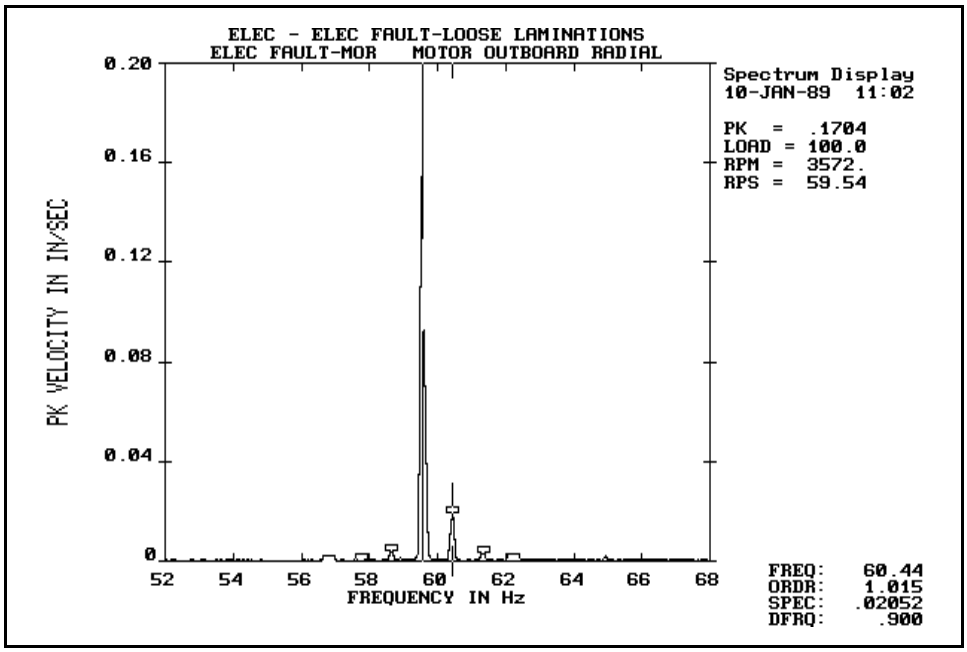

The sidebands around the 1xTS peak should be spaced at a frequency equal to the number of poles on the motor times the slip frequency.

sideband	=	number of motor poles x slip frequency
	=	2 x 0.46 Hz
	=	0.92 Hz

The sideband spacing of 0.9 Hz around 1xTS of the motor approximates this frequency.

Finding these sidebands might cause you to suspect a defective rotor bar.

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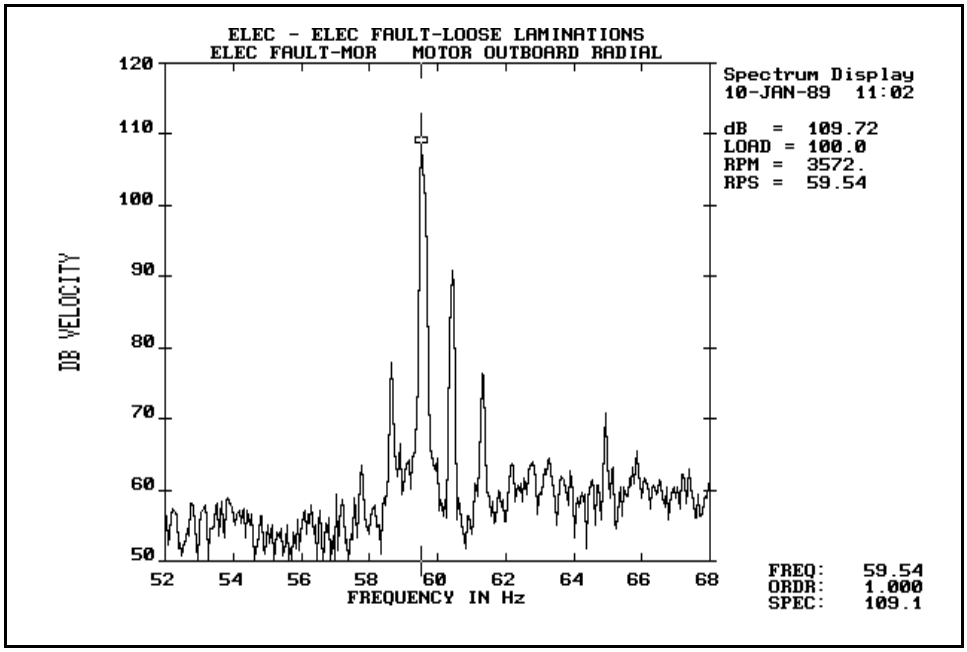

The same portion of the spectrum on the previous page is now plotted in units of dB velocity.

When the sidebands caused by the possible rotor defect are less than 20 dB down from the actual run speed peak, then a significant problem may exist.

Further work with a current probe is needed to examine the electrical characteristics of the motor.

Use the data taken with vibration transducers as a first step in determining whether the machine has an electrical or mechanical problem.

In this case, the motor appears to have an electrical problem.

Note: To change amplitude to dB's, from diagnostic plotting, go to "Global Options", Engr. Units, change Units Mode to desired dB units.

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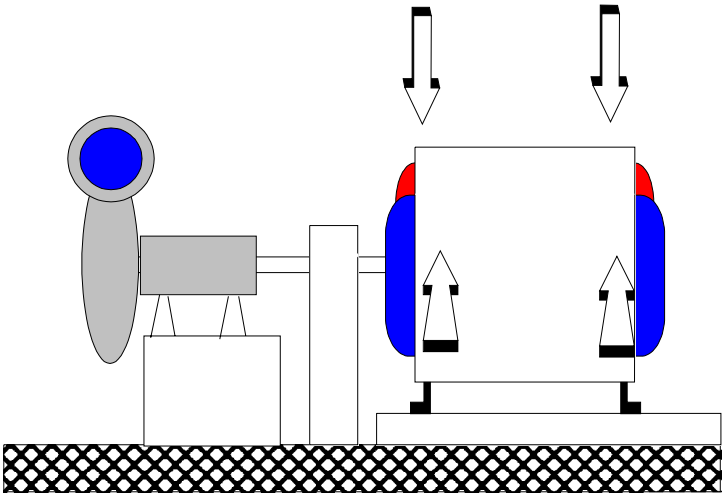
Process Chiller Pump
Electrical Defect Case #3



❶ 15 HP, 1800 RPM AC Induction Motor.

❷ High motor vibration on the motor horizontal points.

❸ Motor vibration at 4.1xTS.

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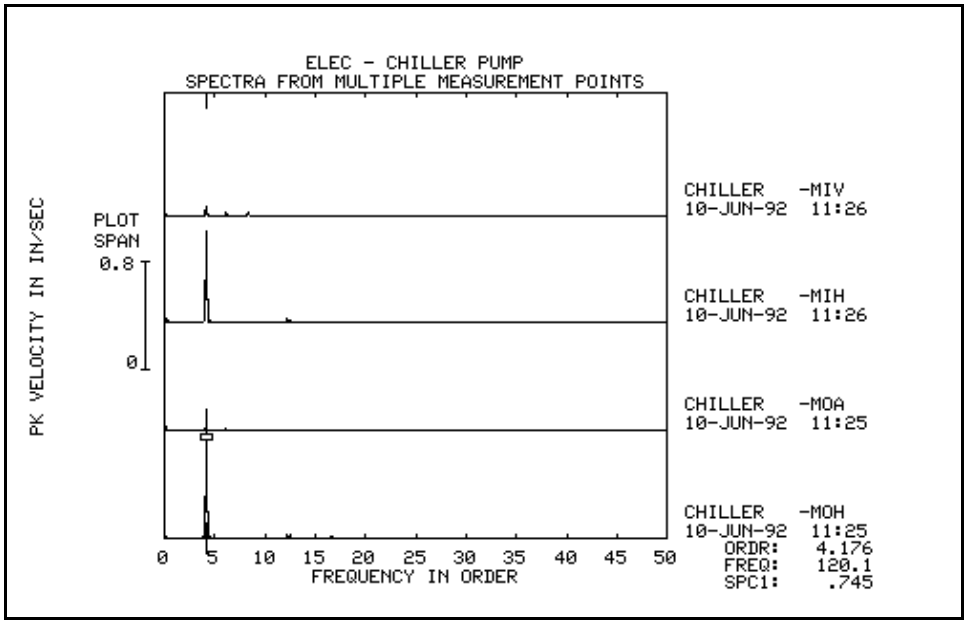
Process Chiller Pump
Electrical Defect Case #3



The multiple plot above comes from a case history involving a direct-driven chiller pump.

The unit running speed peak at 28.75 Hz (1725 RPM) is not even seen in the plots.

Note, the dominant nonsynchronous peak at **4.17 orders** in the horizontal direction.

The spectrum for the motor inboard horizontal appears on the next page.

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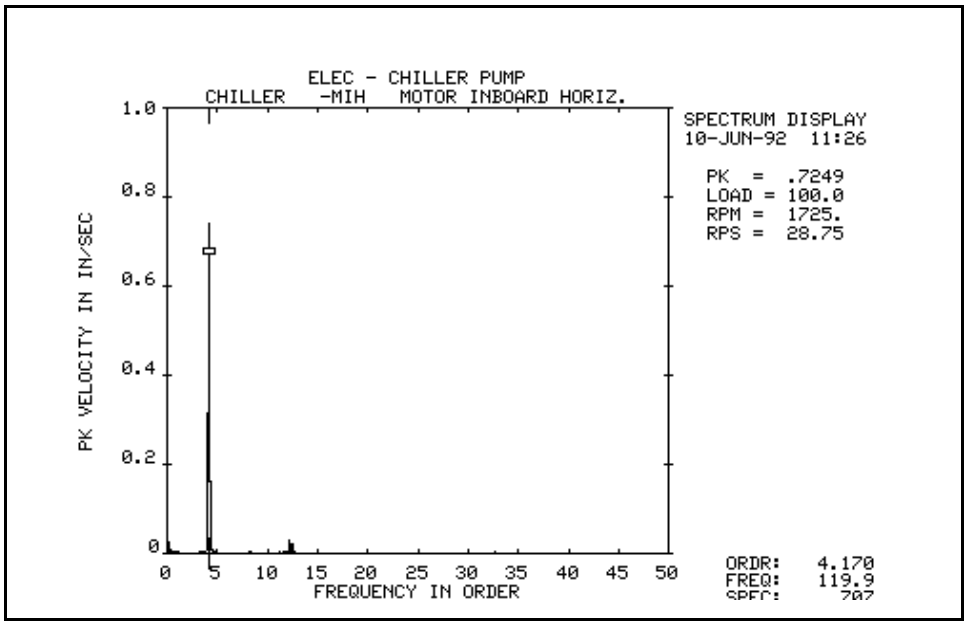
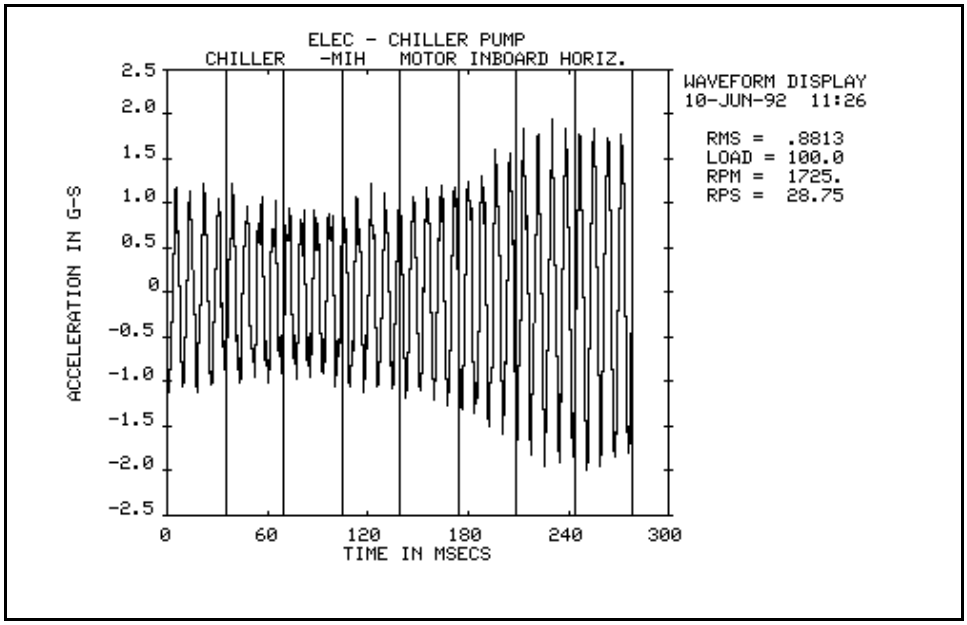
Electrical Defect Case #3





The amplitude of the 4.17 order peak in the spectrum is 0.7 IPS. The waveform shows some modulation and a definite sinusoidal pattern related to the vibration. What could be the source of this directional, nonsynchronous vibration? The next page shows a high resolution spectrum that confirms the suspected problem.

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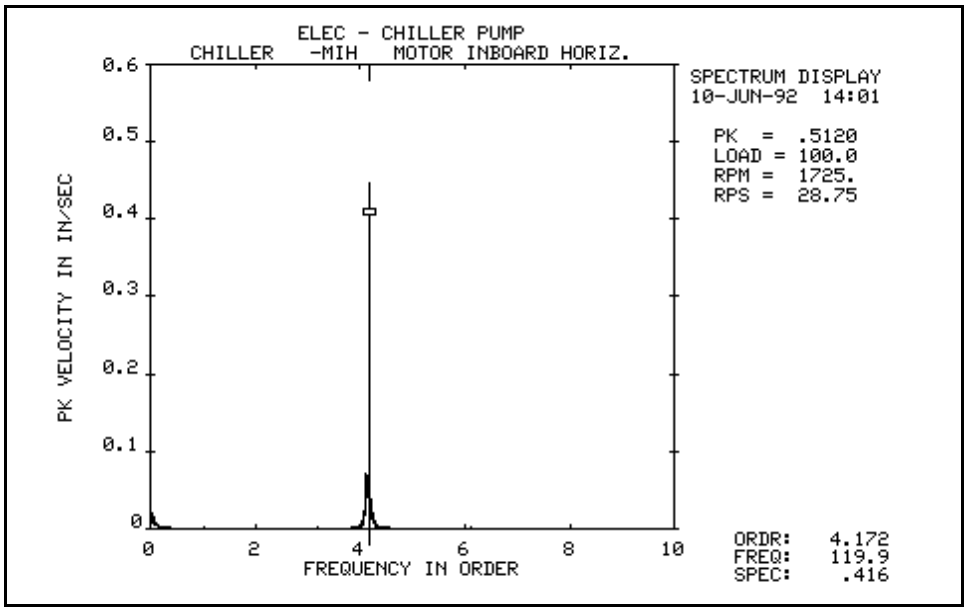
Process Chiller Pump
Electrical Defect Case #3



The located spectral peak is at 4.17 orders or 119.9 Hz.

The vibration at 119.9 Hz (120 Hz) or 2xLF is the problem that needs to be corrected.

Upon investigation, it was found that the directional 2xLF vibration was caused by a soft foot condition on the motor.

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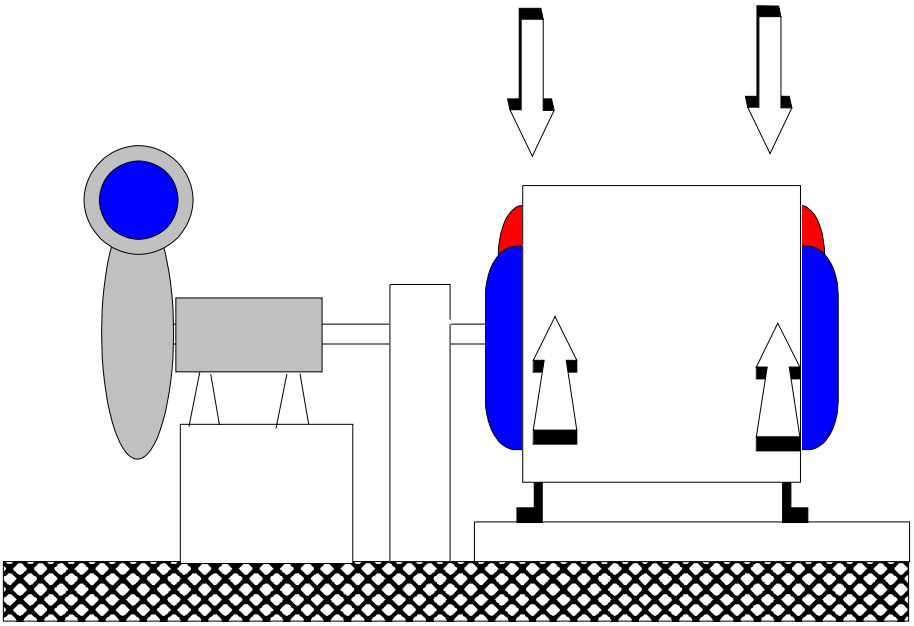
Fan Pump Motor
Electrical Defect Case #4



❶ 450 HP, 1200 RPM Inverter Driven AC Induction Motor.

❷ Motor was running hot and it was replaced.

❸ New motor had the same vibration as the one it replaced.

❹ Current readings were: 93, 95, and 75 amps.

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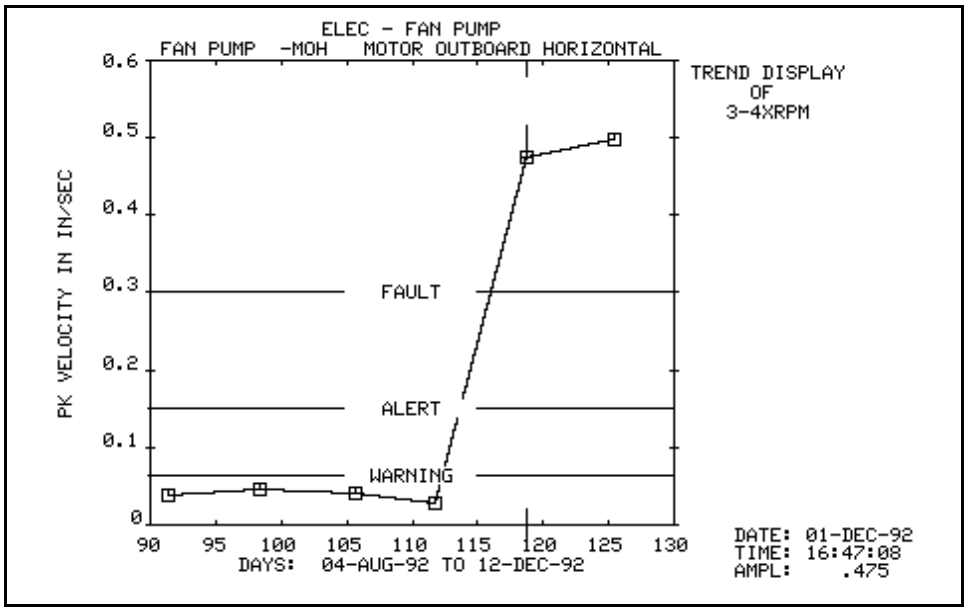
Fan Pump
Electrical Defect Case #4



The trend plot for the motor's 3 to 4xTS band is shown above. This case history is from an inverter driven fan pump.

Notice the large increase in amplitude from the data collected in December.

The next page shows the spectrum that was collected in December.

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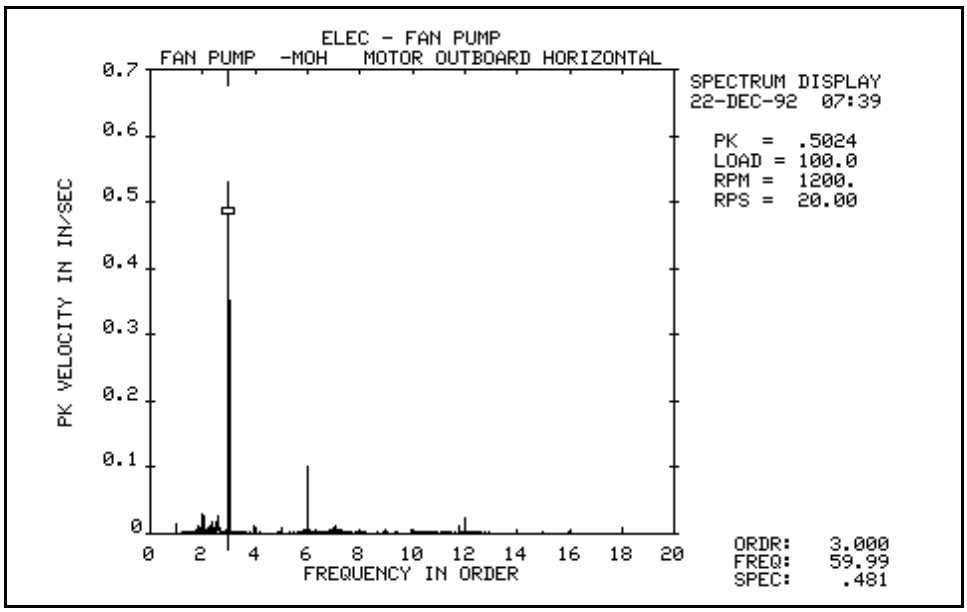
Fan Pump
Electrical Defect Case #4



The trend data clearly shows that something changed in December.

Spectral data from December shows the highest vibration peak at 59.99 Hz.

Setting a harmonic cursor on 59.99 Hz reveals the presence of harmonic peaks.

An electrical defect should be considered since the motor is powered from a variable frequency drive.

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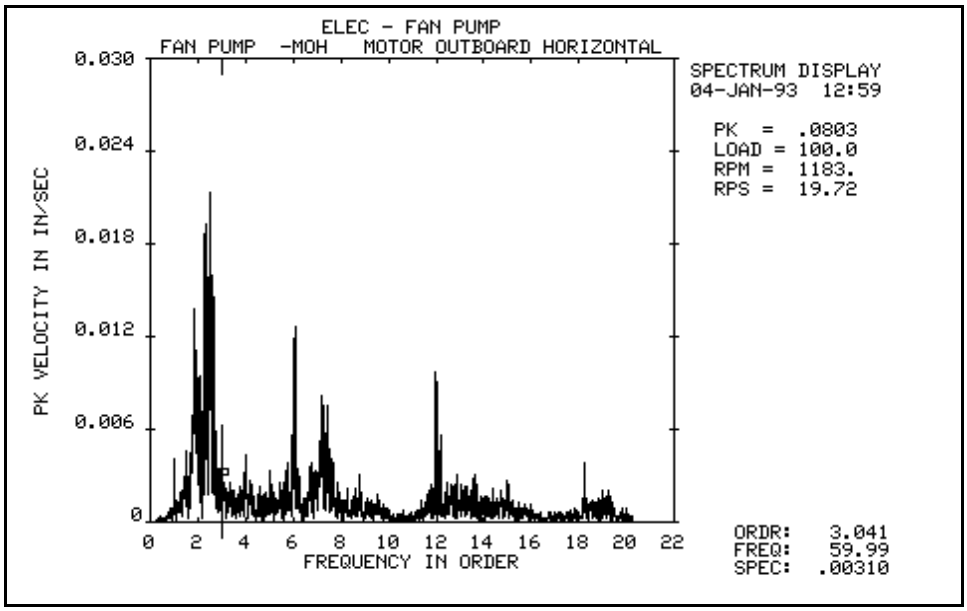
Fan Pump
Electrical Defect Case #4



Troubleshooting was performed to determine the source of the harmonics of line frequency.

This spectrum was taken with one of the boards containing a pair SCRs jumpered out.

Notice that the amplitude of the 59.99 Hz peak went from 0.48 IPS to 0.003 IPS. It was decided that a defective SCR in the variable frequency drive was the source of the 1xLF vibration.

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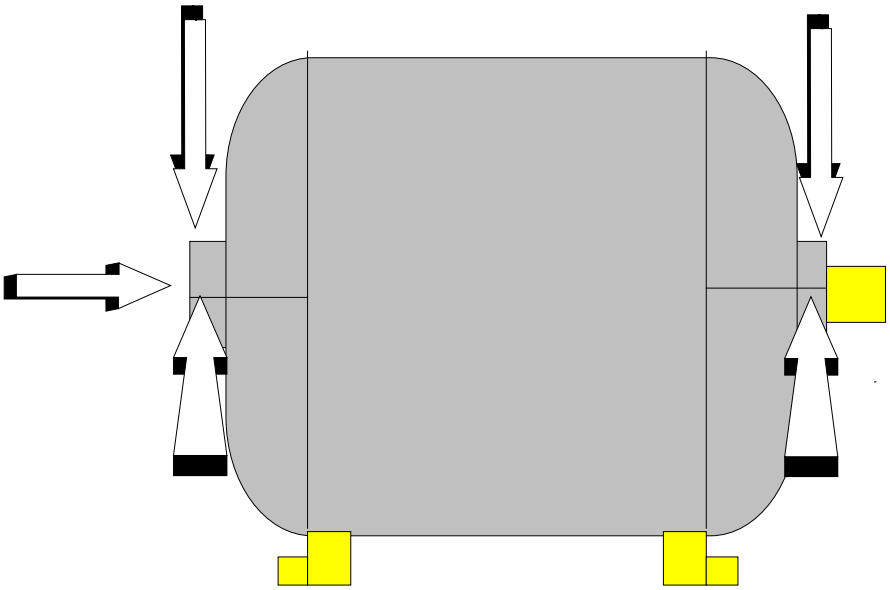
Synchronous Band Motor Electrical Defect Case #5



❶ 500 HP, 900 RPM AC synchronous motor.

❷ The motor was sent in for repair due to high vibration.

❸ High vibration was measured at 2xTS.

❹ The 2xTS vibration was primarily in the horizontal direction.

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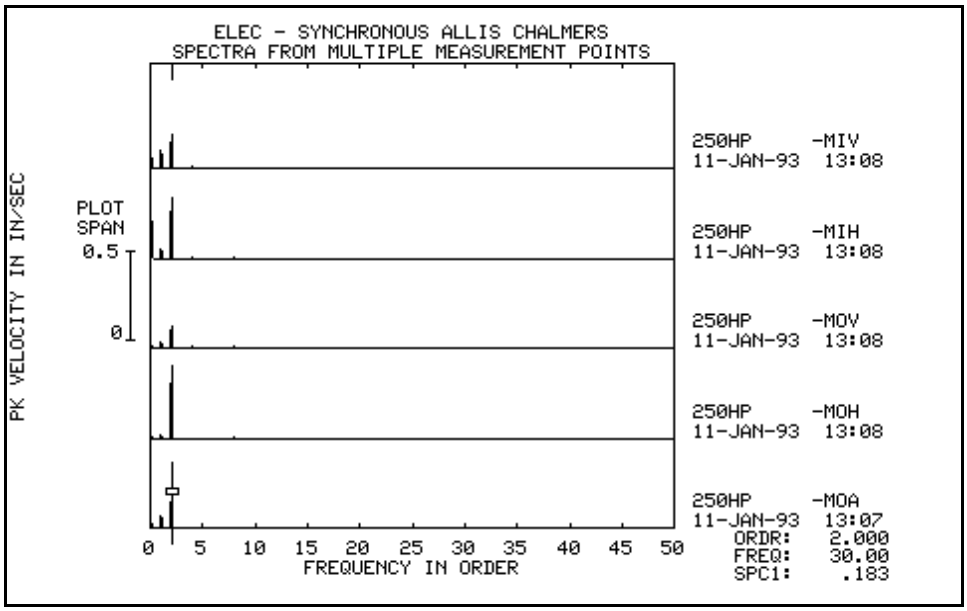
Synchronous Band Motor Electrical Defect Case #5



The multiple plot above shows data from the synchronous motor before repairs were performed.

All the points on the motor show a dominant peak at 2xTS. Troubleshooting was begun to determine the source of the 2xTS vibration.

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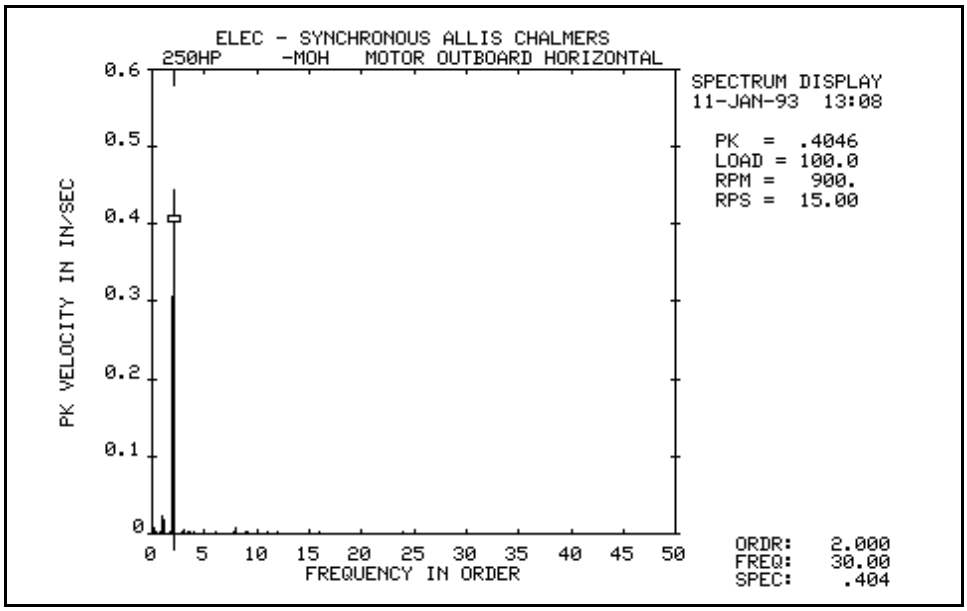
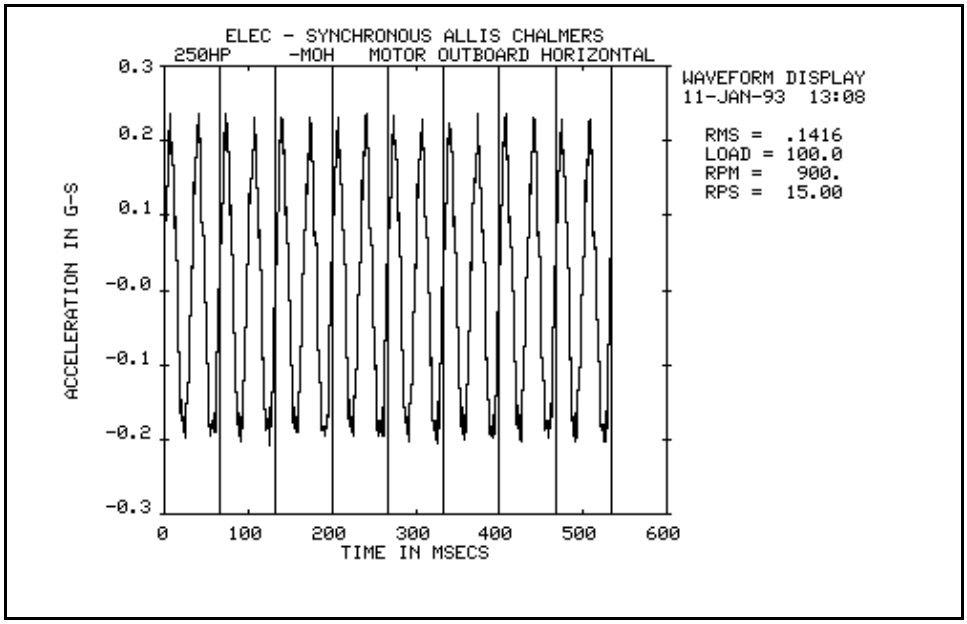
Synchronous Band Motor Electrical Defect Case #5





The Motor Outboard Horizontal (MOH) spectrum and waveform are shown above. Spectral data shows the dominant 2xTS vibration with no harmonics. As expected, the waveform is repetitive and lacks complexity. Notice the low amplitude level in the waveform.

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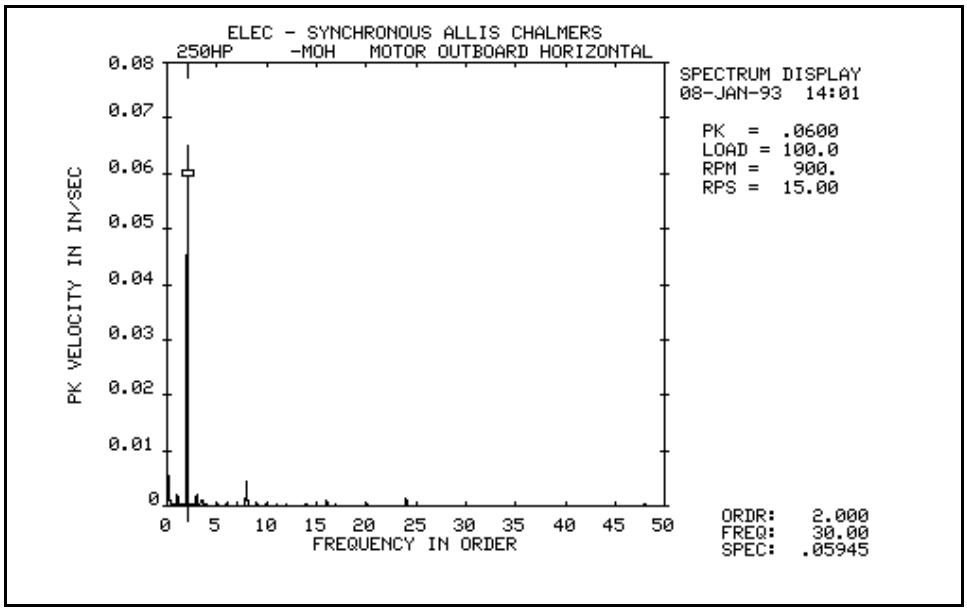
Synchronous Band Motor Electrical Defect Case #5



After extensive troubleshooting, the defect was isolated to the rotor.

Individual rotor pole resistances were taken and some variances were discovered.

The spectrum on this page shows the MOH point after the rotor was rewound. Notice the large reduction in the vibration at 2xTS.

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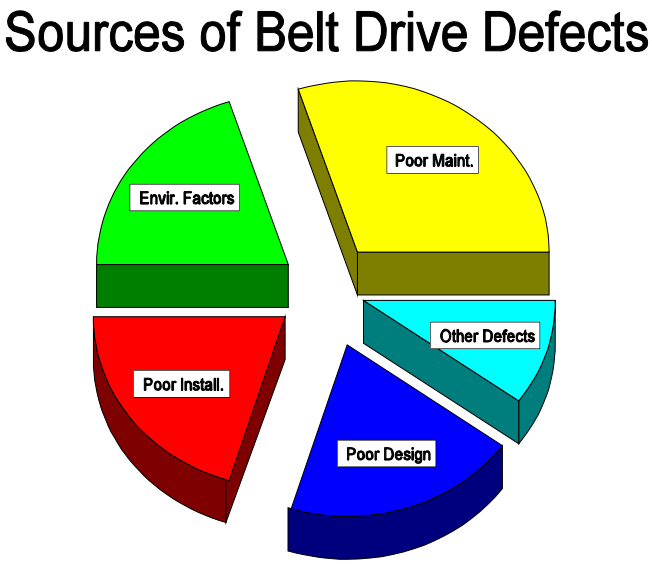
Section Fourteen
Belts

Objectives

- List four causes of belt vibration.
- Describe the vibration characteristics of each belt defect.
- Measure the belt vibration in the most effective locations and directions.

Introduction

Belt defects are considered non-critical faults by many maintenance groups due to the relative ease of replacement requiring minimal downtime. However, belt drive system defects can have a significant effect on the reliability of the equipment. Belt defects themselves are a major contributor to the overall vibration environment of the individual components. Excessive belt tension provides additional load to the component bearings and under tensioning leads to operational performance problems. One main factor affecting component and belt reliability and longevity is proper sheave alignment.



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Defect Characteristics

Belt Faults

Belt faults, such as cracks, broken or missing pieces, hard or soft spots, lumps on belt faces, or a crooked belt which has set to a deformed shape, can generate vibration at the turning speed of the belt (1xBelt) and harmonics.

The predominant harmonic is typically the 2xBelt or the harmonic equal to the number of sheaves in the system, i.e., 3xBelt for a 3 sheave system.

The predominant plane is typically radial and in-line with the belts. Due to the length of the belt in relation to the sheaves, the 1xBelt frequency is subsynchronous and often the 2xBelt may be subsynchronous as well.

Severity is judged by the number and amplitude of the harmonics seen in the spectra.

The belt frequency can be calculated from the following equation:

$$\textit{Belt Freq} = (3.142 * \textit{Shaft TS} * \textit{Pulley Pitch Diameter})/(\textit{Belt Length})$$

Note: The Pitch Diameter and the Belt Length must be in the same units.

The belt length may be known or it can be measured or calculated. If it must be calculated, the following equation can be used:

$$\textit{Belt Length} = 2C * 1.57 * (D + d) + (D - d)^2/4C$$

Where,

C	=	center-to-center distance between sheaves
D	=	pitch diameter of the large sheave
d	=	pitch diameter of the small sheave

Note: ***All the length dimensions must be in the same units.***

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A timing belt will have a specific frequency related to the belt frequency and the number of teeth on the belt and the pulley.

Timing Belt Freq = (Belt Freq) * (# Belt Teeth)

or

Timing Belt Freq = (Pulley TS) * (# Pulley Teeth)

Remember, the belt frequency and harmonics appear when the belts are worn, loose or mismatched.

The defect amplitudes may increase and decrease relative to the driver or driven pulley's turning speed.

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Misaligned Sheaves/Sheave Runout

Belt misalignment is one of the primary reasons for reduced belt life.

If the misalignment is greater than 1/16 of an inch for each 12 inches of belt—center to center distance—very rapid wear will develop.

Very often, sheaves are out of line because the motor and the machine shafts are not parallel.

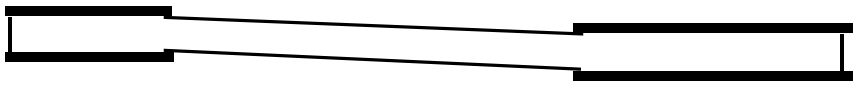
Sheave misalignment results in high **axial vibration** at shaft turning speed. If belt defects are also present, belt defect vibration will also become more evident in the axial plane.

Misaligned sheaves show high vibration at 1xTS vibration predominantly in the axial direction. Whether the driver or driven 1xTS peak is dominant is dependent on the mass and stiffness of each component.

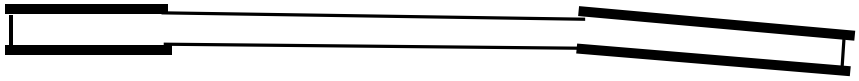
Not always, but often with sheave misalignment the highest axial vibration on the motor will be at the fan turning speed and the highest axial vibration on the driven unit will be at motor turning speed. This is referred to as, **Cross Over** effect. Belts that are adjusted too tight can also cause this situation.

Also 3xTS and 4xTS multiples of the belt frequency may show up in line with the belts and in the axial direction.

Parallel Misalignment



Angular Misalignment



A wobbling sheave whips the belt from side to side subjecting it to excessive lateral strain. This can lead to higher axial 1xTS vibration.

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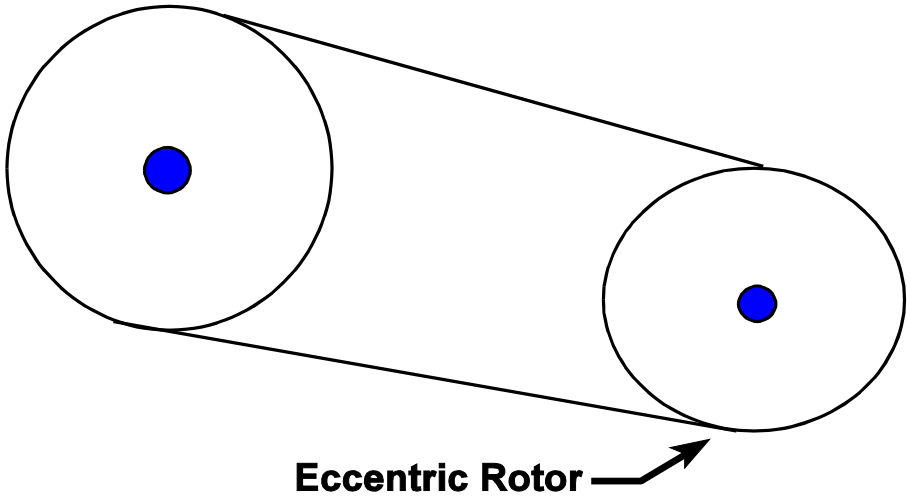
Eccentricity/Imbalance

If the sheave is true but unbalanced, vibration occurs at shaft turning speed radially. This relates to an unbalance problem which will show up at 1xTS frequency.

If the sheave is balanced but has excessive runout, symptoms of inadequate tensioning become evident but at low amplitudes.

Runout also tends to generate excessive belt-flapping and adds to the radial turning speed amplitude of the defective shaft.

High vibration will show up at 1xTS as the sheave literally has a high spot that affects the belt tension once per revolution of the sheave.



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Belt Resonance

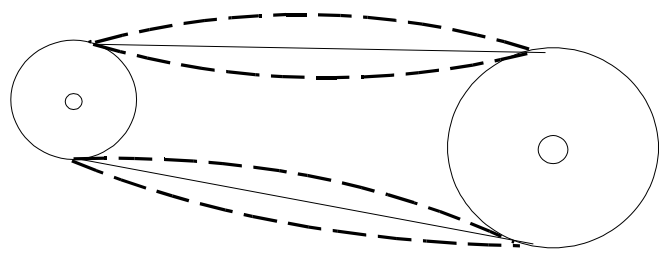
Every system has resonant frequencies and belt systems are no different. There are two types of belt resonance problems.

❶ Belt frequencies exciting other system resonances.

Belt defects can generate many different frequencies that can excite system resonance, such as turning speed and harmonics of both the driver and the driven shaft. Also, belt frequency and multiples of belt frequency. This may be the case when one of the multiples of a belt defect has a high amplitude compared to the other belt frequency harmonics.

❷ Belt Strand Resonance (BSR) being excited by other system frequencies.

The belts also have a system resonance of their own. This frequency is often referred to as the Belt Strand Resonance. The BSR is related to the portion of the belt suspended between the two sheaves. The frequency of the BSR is dependent on the mass, stiffness, damping properties of the belt as well as the belt tension. This is similar to the properties of a violin or a guitar string.



The BSR can typically be determined by an experimental measurement. With the machine off, the belt can be plucked and the resonant frequency can be measured. The analyzer setup will be similar to that for an impact test. The accelerometer can either be taped to the belt or placed at the sheave bearing.

The BSR should be adjusted if it will be excited by one of the operational frequencies of the machine train. Adjustments could be made by altering the belt length, the tension, or by placing an idler pulley in the system to effectively change the belt length.

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Data Collection Considerations

Vibration Data

Due to the proximity of the 2xBelt frequency to the shaft turning speed frequency (1xTS), the ability to separate the two peaks for analysis is extremely important.

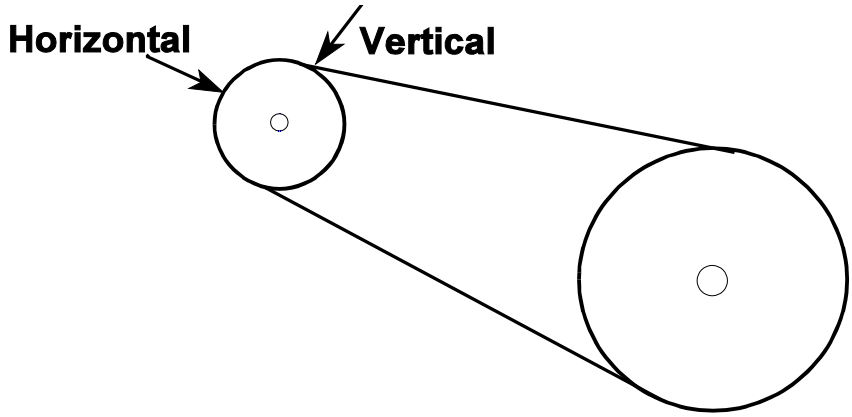
Use a high resolution measurement point on each shaft in the in-line position for effective analysis of belt-driven systems.

For most belt-drive systems, using a normal data collection resolution (800 lines) is usually sufficient with an F_{max} of 3xTS of the high speed shaft.

However, each system should be evaluated individually to ensure adequate separation of belt defect harmonics from other peaks. Caution should be taken with your accelerometer mounting technique to allow the 1xBelt to be viewed without interference from integration noise.

This high-resolution data may be collected in addition to normal route-collected data in the analyze mode based on field observations. The most effective method for trending and fault progression analysis is to make these measurements part of your normal route data collection.

The best measurement directions to detect belt defects and to see the highest amplitude vibration is in line with the belt tension (horizontal) and perpendicular to the belt tension (vertical).



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Physical Observations

An extremely effective technique for analyzing belt system faults is to use a strobe light in conjunction with your data collection.

Belt defects are often visible when the belts are frozen.

Sheave misalignment is also typically visible when the sheaves are frozen.

Applying a reference line across the belts during installation will provide a comparison point for determining the amount of slippage in the system.

Summary

This section has covered the basic belt defect causes and the frequencies that can be generated from these frequencies.

Defects related to problems with the belt itself can generate 1xBelt and harmonics.

2xBelt can be dominant because of the two sheaves that a belt defect must pass over.

Other problems, such as belt misalignment and sheave runout can lead to a higher 1xTS in the axial direction.

A sheave can be well balanced but still generate a 1xTS in the radial direction due to the sheave being eccentric (out of round). This causes uneven belt tension effects once per revolution.

The belt strand resonance can be a significant problem that is often overlooked. Generally, the BSR must be measured in order to know if it can be excited by some other system operating frequency. A belt drive system may rapidly fail if this frequency is excited.

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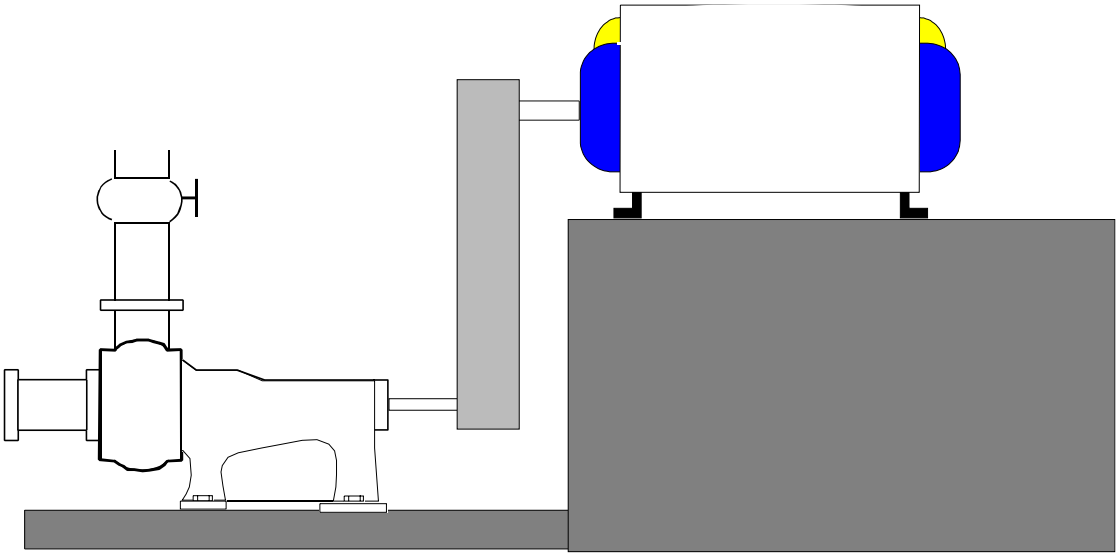
Case Histories

Belt Case 1



This case history typifies data for misaligned sheaves. The data shows a severe condition affecting the motor measurement points.

Allowing this condition to progress would lead to premature bearing failure.

Identification and correction of this condition increases both the reliability and efficiency of the motor.

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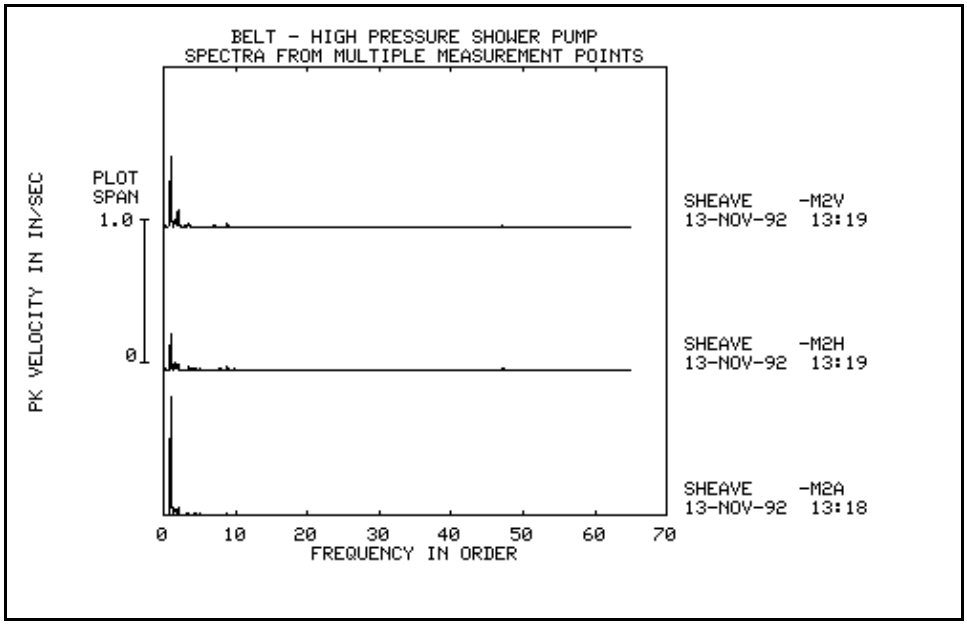
Belt Case 1



The above multiple-spectra plot shows data from the Motor Inboard measurement points.

Notice the highest amplitude appears at 1xTS and the predominant plane is the axial reading.

Very little vibration is seen at harmonics of turning speed indicating that this problem is in its early stages and has not caused any looseness to develop.

With a plot span of 1.0 in/sec, it is difficult to determine whether bearing defects have begun to appear as a result of the misalignment condition.

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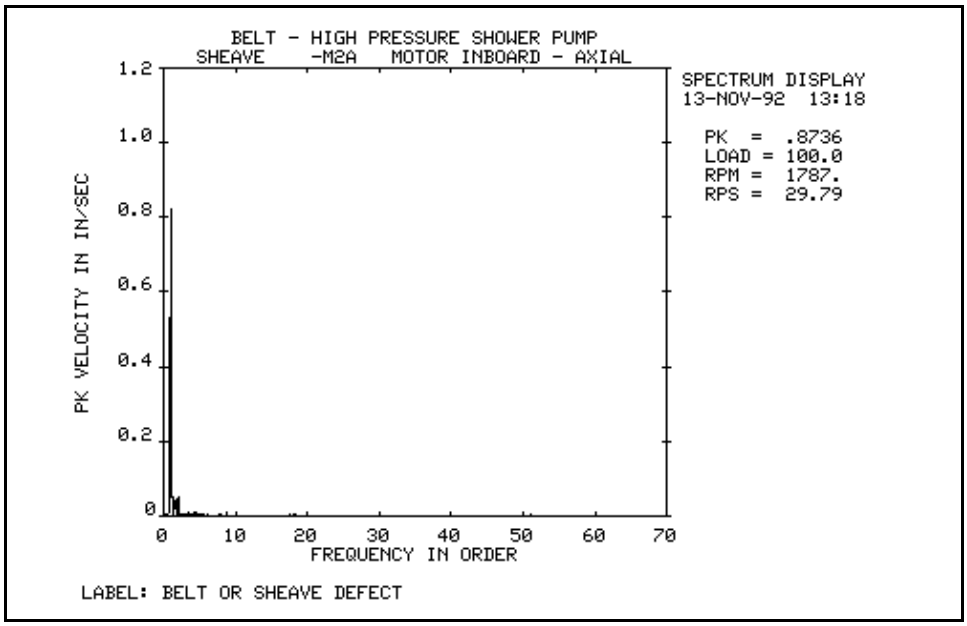
Belt Case 1



The spectrum for the Motor Inboard Axial (M2A) is shown above.

Most of the overall vibration is due to the energy located at **1xTS** and further analysis would be required to determine whether other problems have been driven by this condition.

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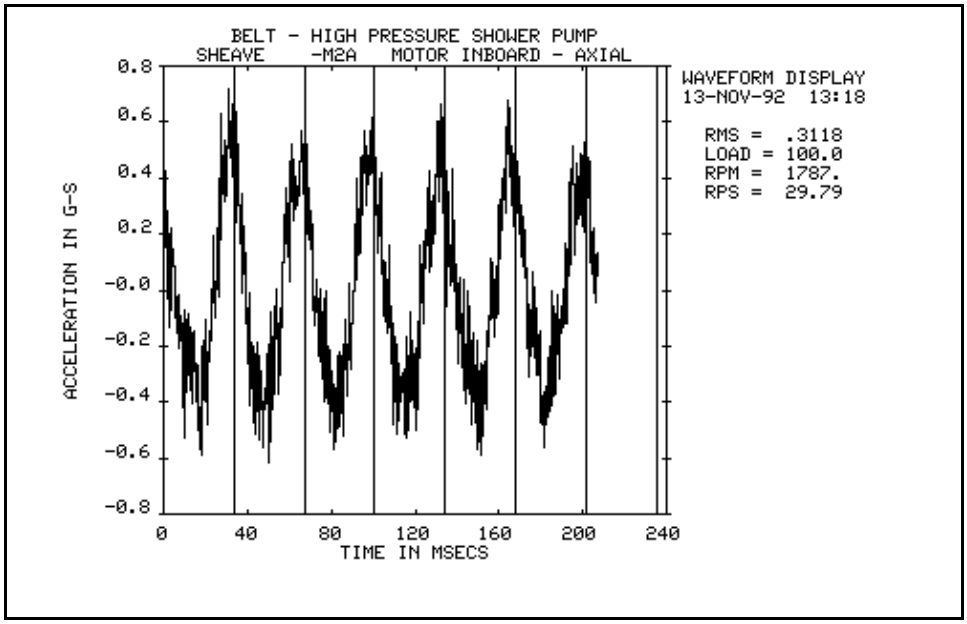
Belt Case 1



As suspected, the waveform is showing a fairly nonimpacting, sinusoidal pattern with one event per cycle. The amplitude is fairly high and helps verify the source of the 1xTS vibration as misaligned sheaves.

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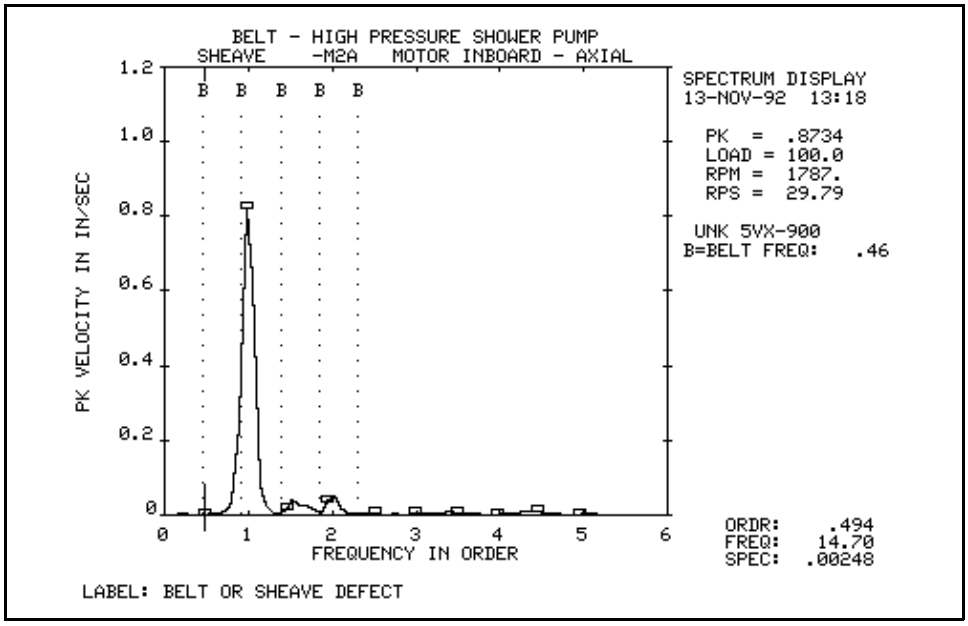
Belt Case 1



Expanding the frequency axis allows us to determine the condition of the belts.

The fault frequencies are displayed for the primary belt frequency with harmonics.

Notice the primary belt defect frequency (1xBelt) does not appear in the spectrum, possibly due to the amplitude scale.

The 2xBelt frequency may be adding to the 1xTS peak amplitude. Notice that the skirt width of the 1xTS peak includes the frequency for the 2xBelt.

The occurrence of the wide-skirted peak should cause the analyst to consider whether the peak is made up of two or more frequencies added together.

Obtaining high resolution spectral data would help to separate the 1xTS from the 2xBelt frequency and allow us to determine the condition of the belts.

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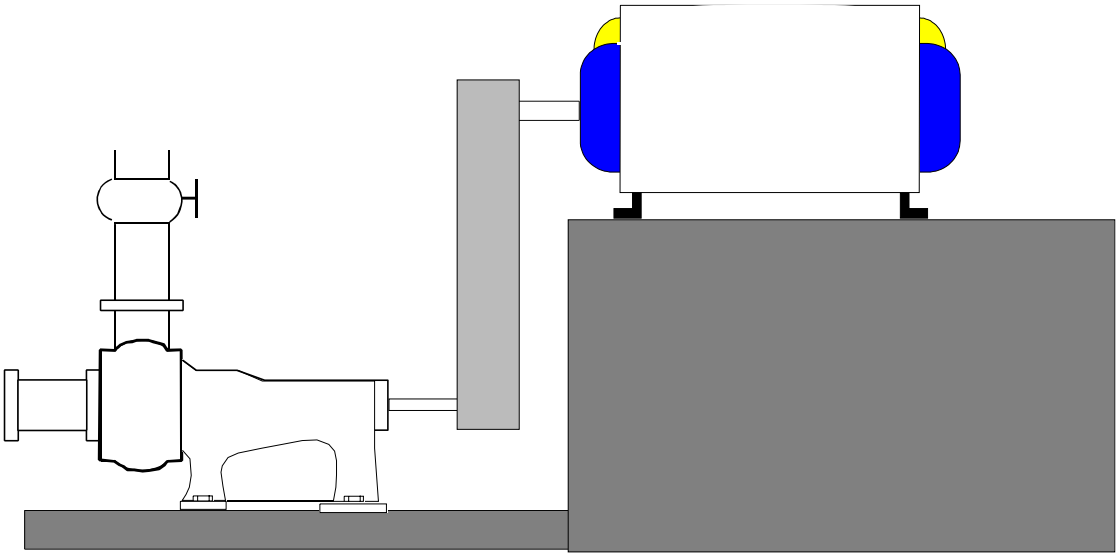
Belt Case 2

This case history examines the need for resolution for defect analysis and the identification of the need for additional data during normal route data collection.



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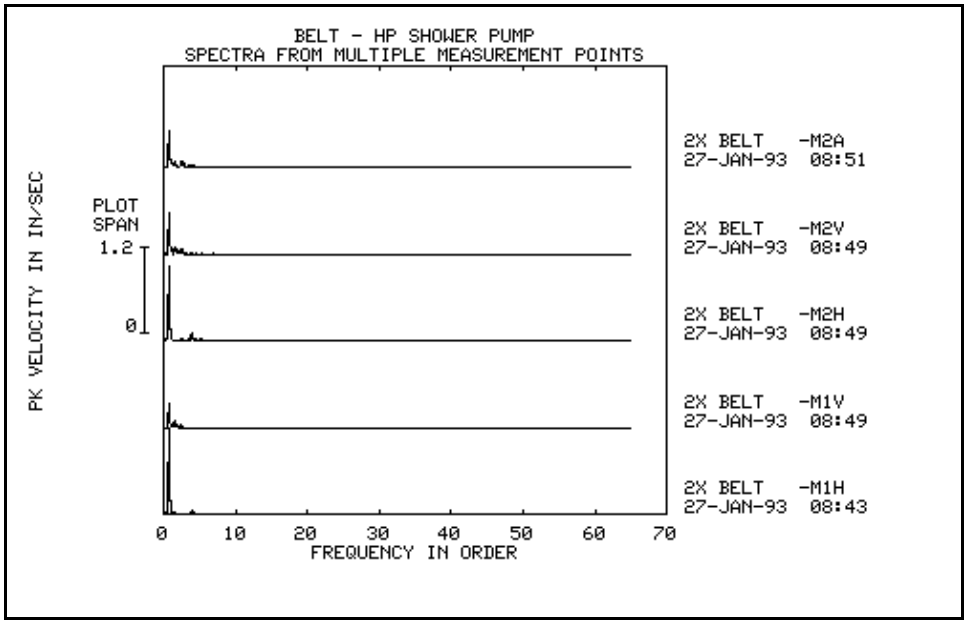
Belt Case 2



The multiple-point spectral plots show route-collected data for the measurement points on the motor.

Notice most of the energy appears to be around 1xTS with the radial points showing the higher amplitudes.

Measurement point M1H, Motor Outboard Horizontal, appears to have the most energy and is displayed on the next page.

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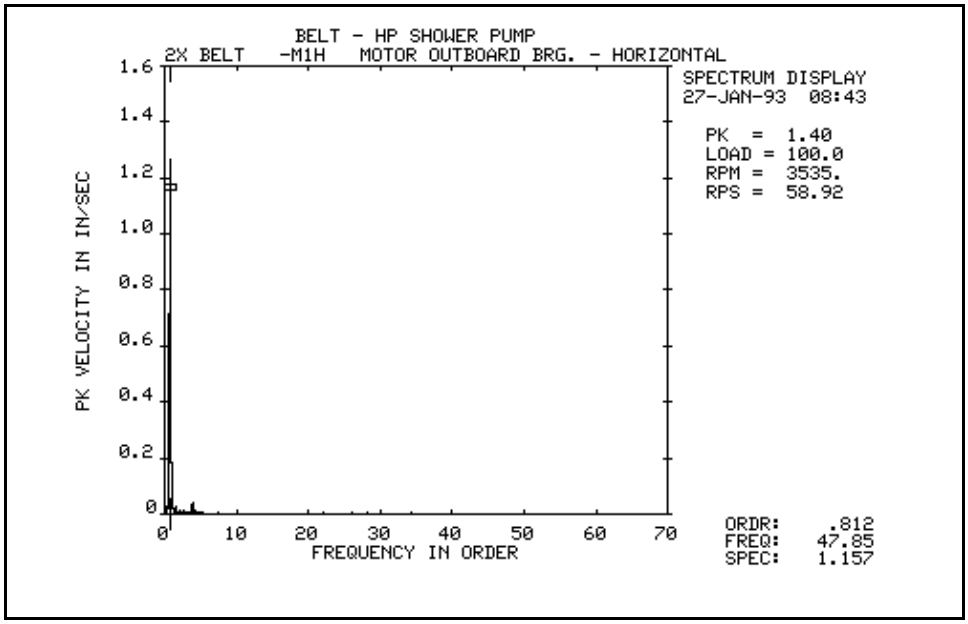
Belt Case 2



The full-screen view of M1H autoranges to 1.6 in/sec and does not provide much detail of the lower amplitude frequencies.

Viewing this data in the field may lead you to consider the high amplitude peak as turning speed.

The cursor on the main peak identifies the peak frequency is actually about 0.8xTS.

The person collecting the data recognized this may be due to a belt defect and acquired additional data as shown on the next page.

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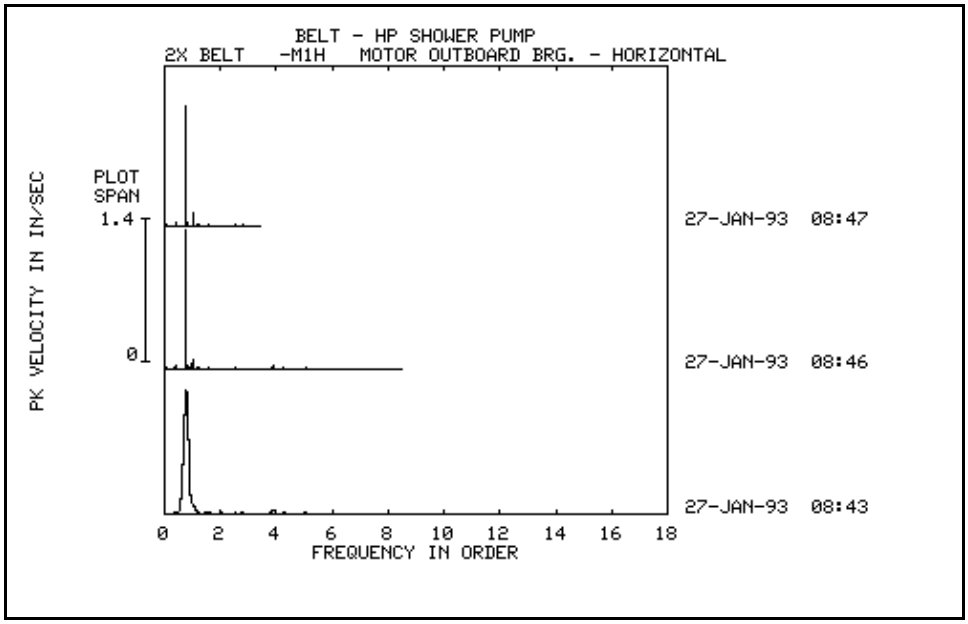
Belt Case 2



Notice the wide-skirted peak in the spectrum from the route spectral setup at 08:43.

The 2xBelt frequency and 1xTS cannot be separated in this spectrum.

The analyst suspected a belt problem and acquired additional high-resolution spectra with a lower F_{max} and higher lines of resolution (LR).

The spectrum from 08:46 had an F_{max} of 500 Hz and 3200 LR.

The spectrum from 08:47 had an F_{max} of 200 Hz and 3200 LR.

The following pages show the analysis of the high-resolution spectrum from 08:47.

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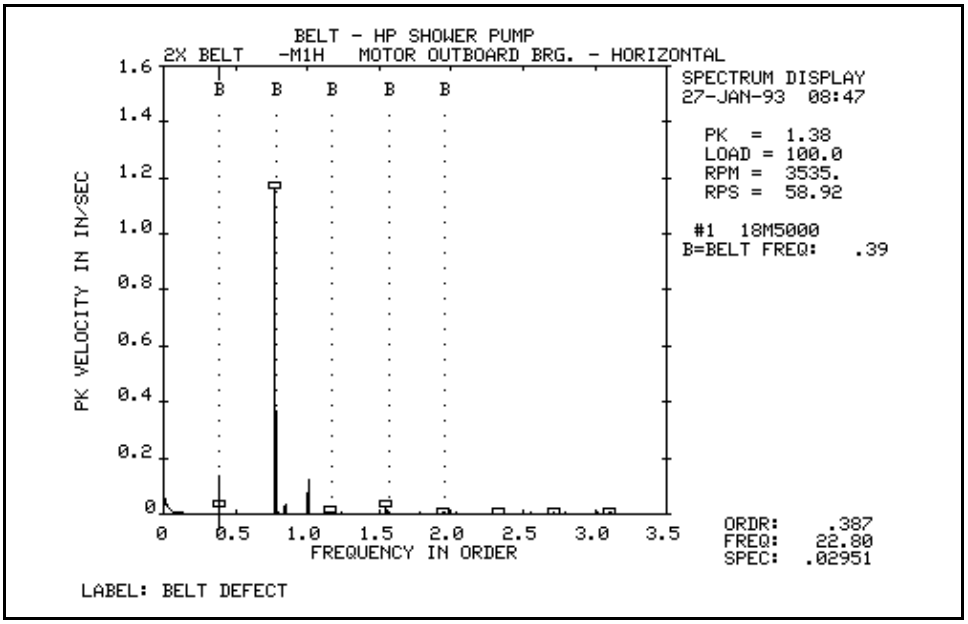
Belt Case 2



The spectrum above displays the belt defect fault frequencies for the belt defect.

Notice the 2xBelt frequency is the dominating peak in the spectrum.

The overall value for the spectrum is 1.38 in/sec and the amplitude of the 2xBelt frequency is 1.2 in/sec.

Most of the overall energy is a result of the 2xBelt frequency, not 1xTS.

Obtaining this data in the normal route mode would simplify the data collection process and allow this fault to be accurately identified during the data collection process.

Having trend data of this machine would then allow the analyst to determine the rate of progression more accurately and provide valuable information for determining the recommended maintenance time frame.

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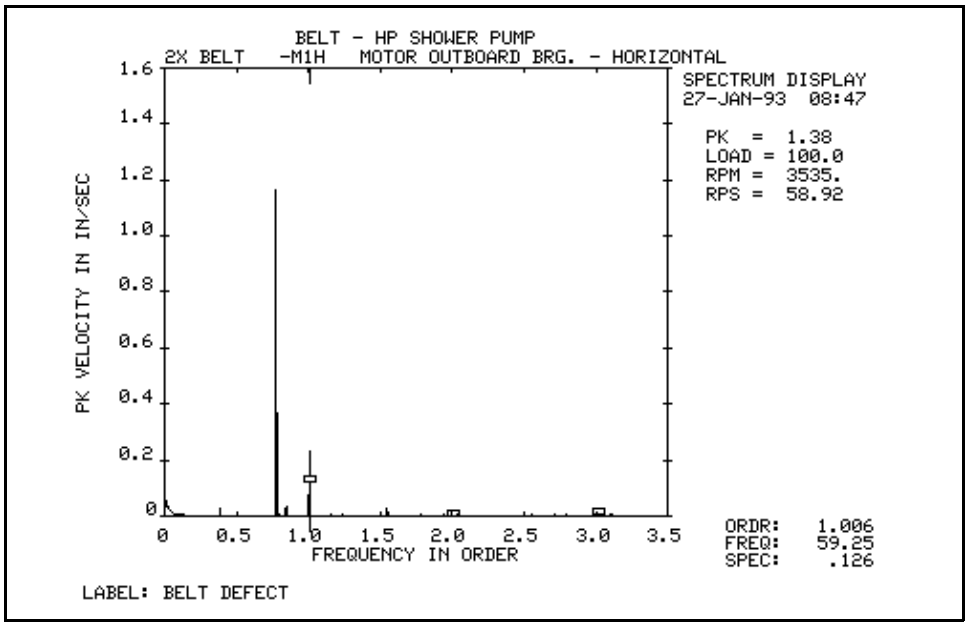
Belt Case 2



The cursor marks the location of the 1xTS peak.

Without the resolution available in this spectrum, the 1xTS and 2xBelt frequency may actually appear as the same peak.

The belt problem may have been incorrectly diagnosed as unbalance when the 1xTS amplitude was actually acceptable.

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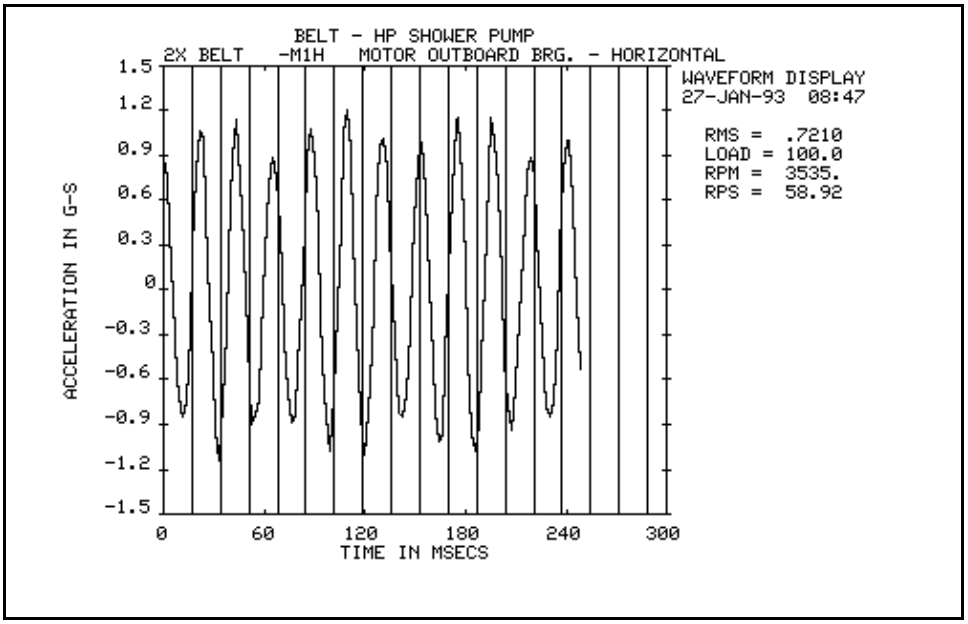
Belt Case 2



As expected, the associated time waveform shown above shows a strong sinusoidal pattern.

At first look, it appears to have a regular one event per cycle.

Notice that the events do not occur at regular intervals because the predominant vibration is due to the 2xBelt.

There also appears to be some modulation resulting from the minor amount of vibration at 1xTS.

The modulation occurs at a frequency equal to the difference between 1xTS and 2xBelt.

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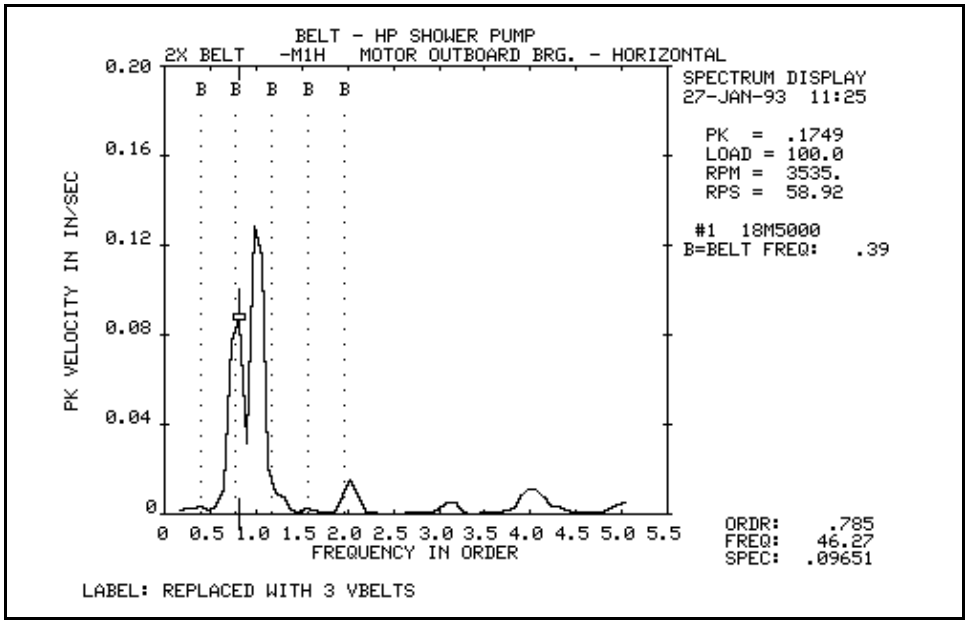
Belt Case 2



The spectrum shown above was collected after the belts were replaced.

Notice the amplitude of the 2xBelt frequency is still fairly high but has dropped to about 0.09 in/sec and the overall level has decreased to 0.1749 in/sec.

Although the immediate problem was corrected, 0.09 in/sec at 2xBelt should not be considered acceptable.

This case history not only allows us to identify that a defect exists, but also contains a probable cause for the belt defect.

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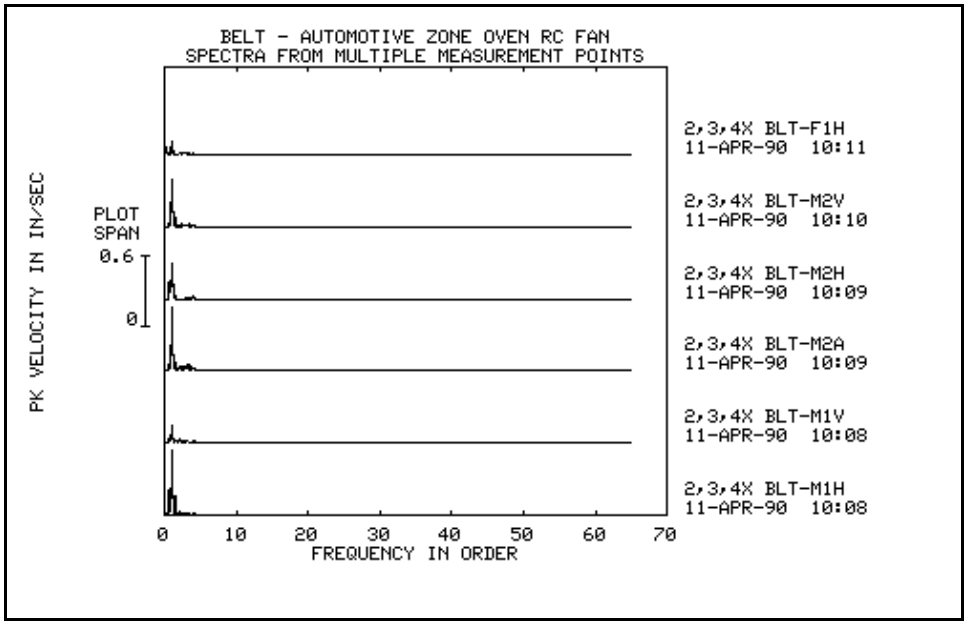
Belt Case 3

The multiple-spectra plot shown below indicates that all measurement points have skirted peaks at what appears to be turning speed.

The Motor Inboard Axial (M2A) measurement point appears to have the highest amplitudes and we will examine it further.



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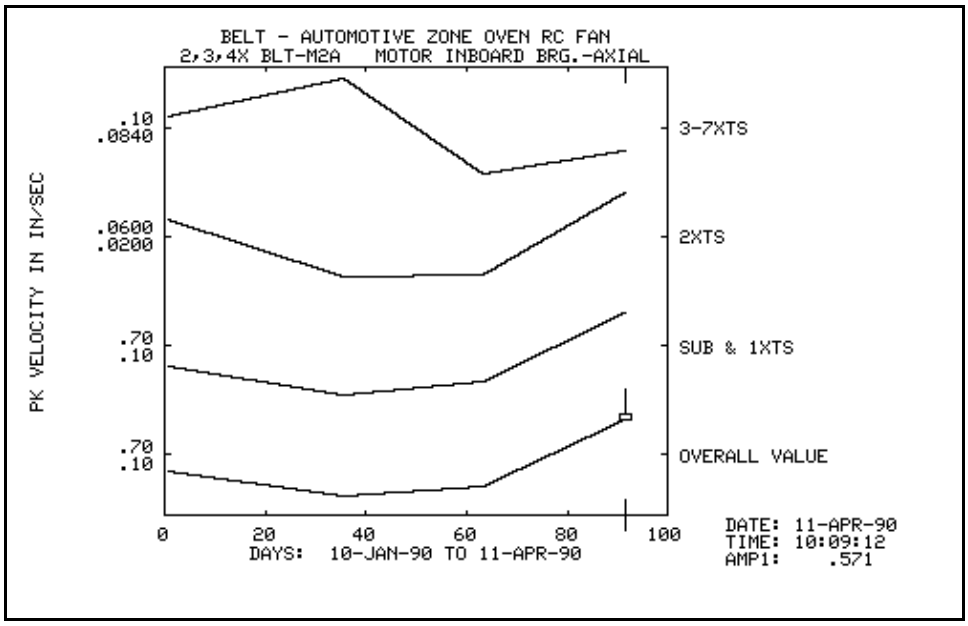
Belt Case 3



The multiple-trend plot from point M2A shows that the overall amplitude has increased significantly over the last reading.

The trend of the SUB and 1xTS band has seen the same amplitude increase. The 2xTS band has increased only slightly.

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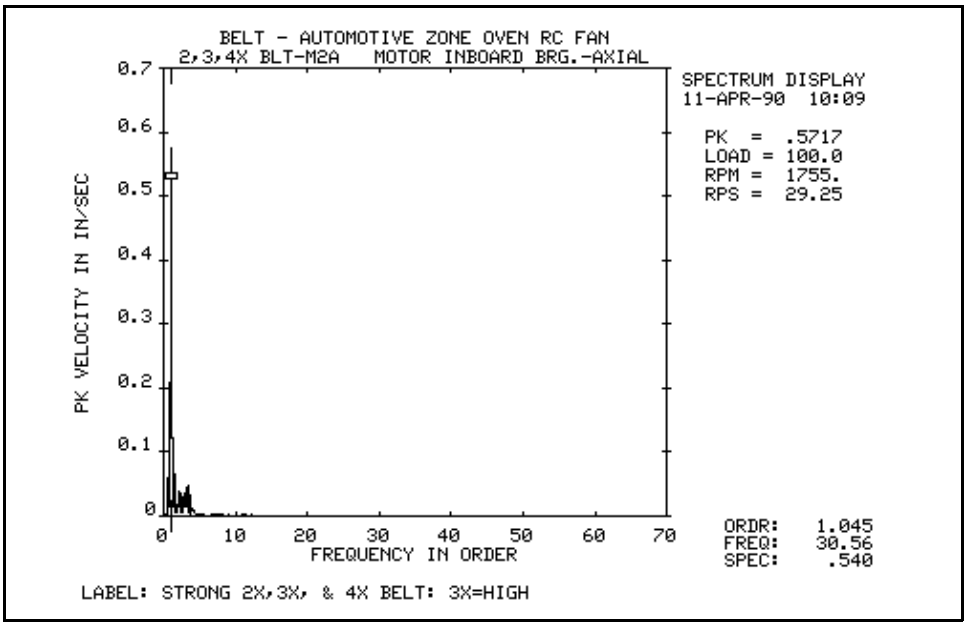
Belt Case 3



The single spectrum displayed above shows a mound of energy around 1xTS.

The frequency scale needs to be expanded to see what may be occurring there.

Note that the low amplitudes of the other frequencies in the spectrum may be due to the amplitude scale (0-0.7 in/sec).

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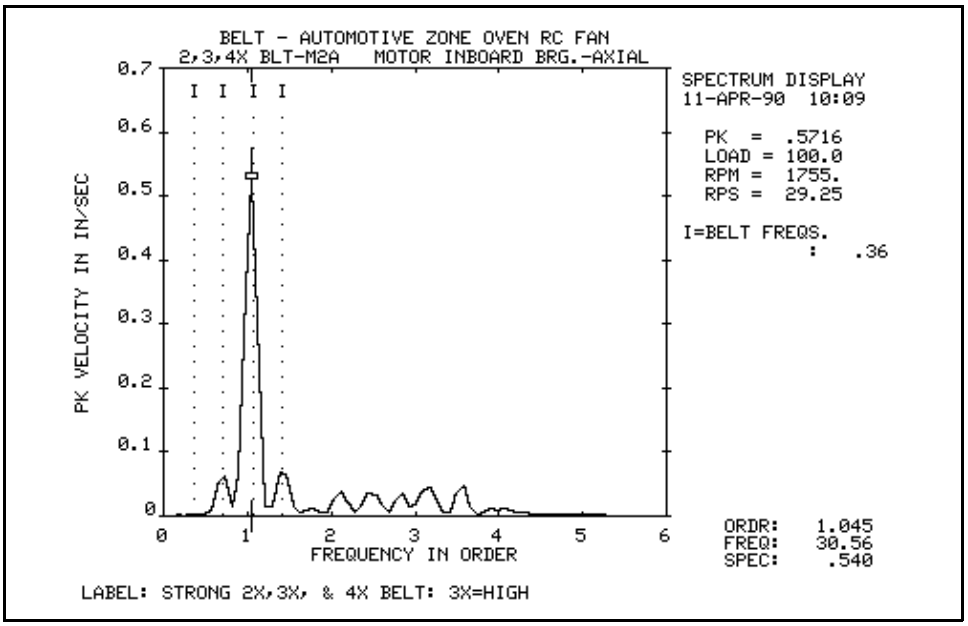
Belt Case 3



The spectrum displayed above is an expanded view of the plot around 1xTS.

The fault frequencies are displayed for the belt defect.

Notice the 2xBelt and 4xBelt frequencies are clearly seen in the spectrum. However, the 1xTS frequency is the dominating frequency.

As with the previous case, the 1xTS skirt width may provide an indication that there are two closely spaced peaks appearing as a single peak.

This spectrum does not have enough resolution to separate the two peaks.

The dominance of the 1xTS peak in the axial plane proves this is a misaligned sheave problem and the harmonics of the belt defect frequency axially indicate that the belts are suffering from the alignment condition.

The sheave misalignment has damaged the belts.

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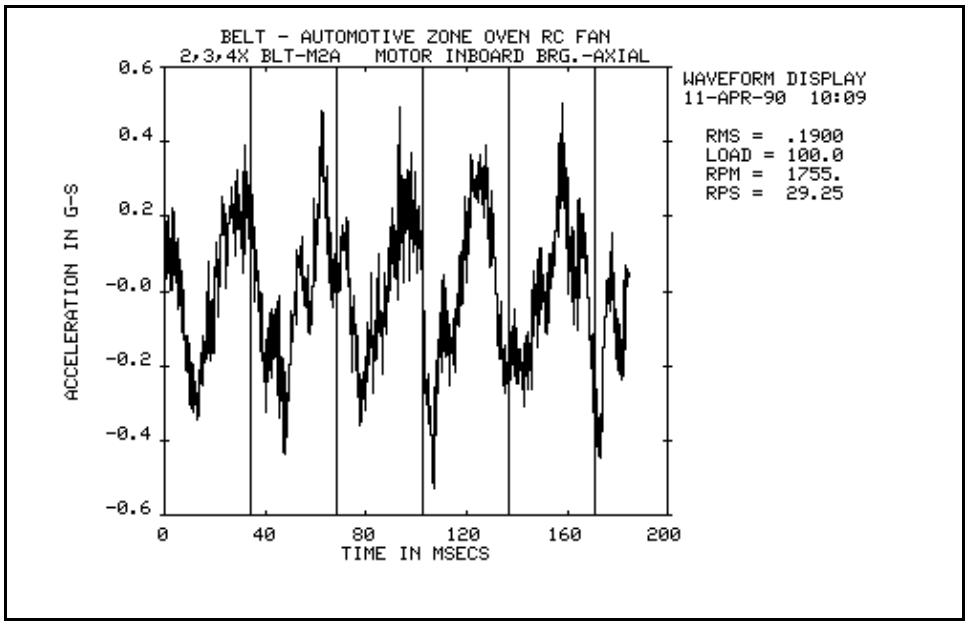
Belt Case 3



The waveform shows one event per cycle as expected. However, the time span is limited and we are unable to identify any modulation due to the belt fault.

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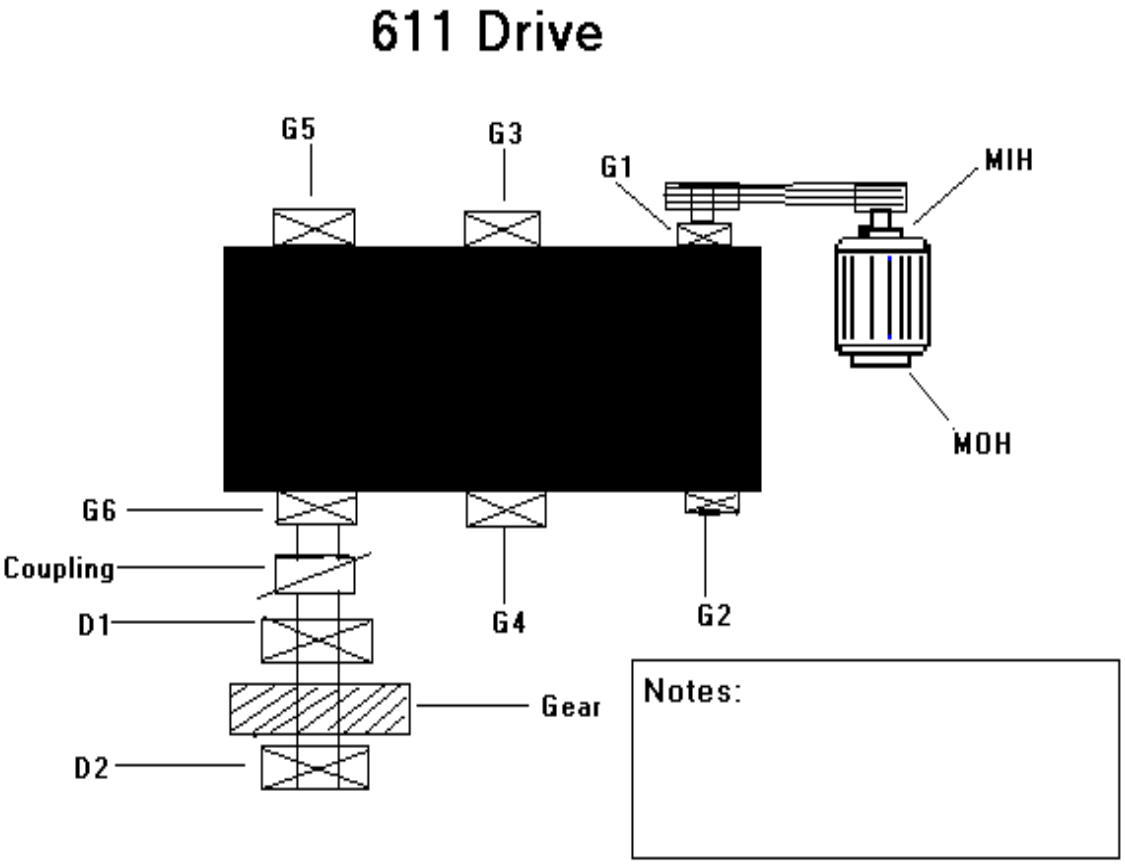
Belt Case 4



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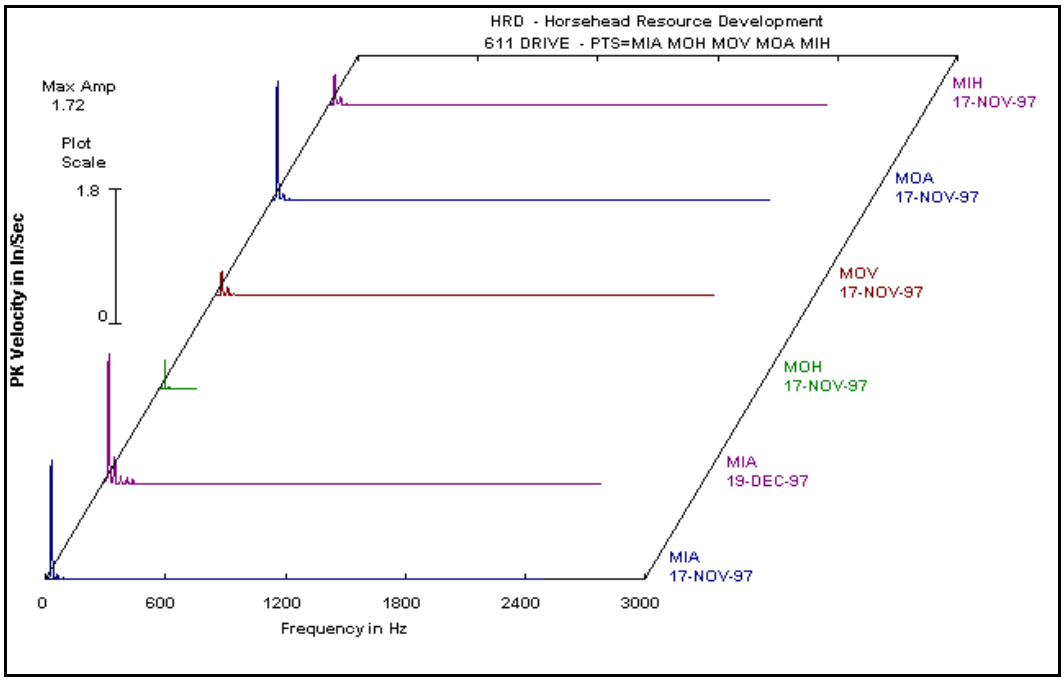
Belt Case 4

In this multiple points plot we can see the amplitude of the vibration in the Axial direction.



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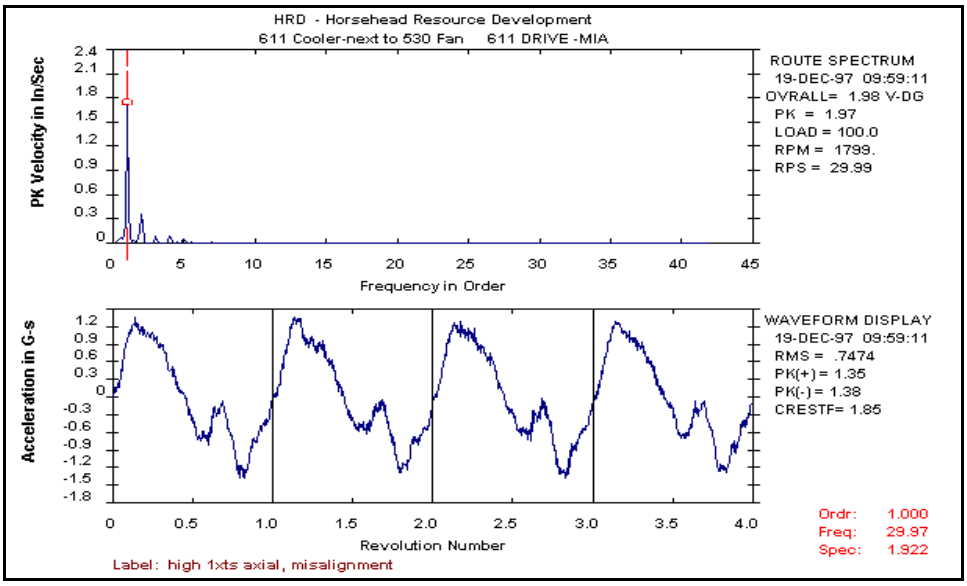
Belt Case 4

Note the high amplitude at 1 x Turning Speed, 1.97 in/sec in the axial direction. Data collected 19th Dec 1997.



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Belt Case 4

Compile report was written as follows:

Case History Data Entry - PD01

Horsehead Resource Development — 611 Cooler-next to 530 Fan

Machine ID: 611 DRIVE — Saved Plots: 1

Survey Date: 19-DEC-97 — Analyst: REC

Problem Desc: POSSIBLE SHEAVE MISALIGNMENT, LOOSE BELTS/defects

Findings : VERY HIGH 1X TURN SPEED VIBRATION AXIAL DIRECTION INDICATION OF BELT/SHEAVE MISALIGNMENT

Recommendations : CHECK SHEAVES WITH STRAIGHT EDGE, TENSION BELTS PROPERLY, CHECK SHEAVES FOR WEAR.

Priority : 3 — Date of Work Order: 19-DEC-97

Fault Type: 6 — Work Order Number:

Equip Type: 18 — Job Complete Yet?: ☐

Current Status of Job: Pending/REPORTED TO MAINTENANCE

Date of Follow-Up :

Follow-Up Actions and Results : Sheave was replaced on the motor. New belts installed and straight edge sheave alignment performed.

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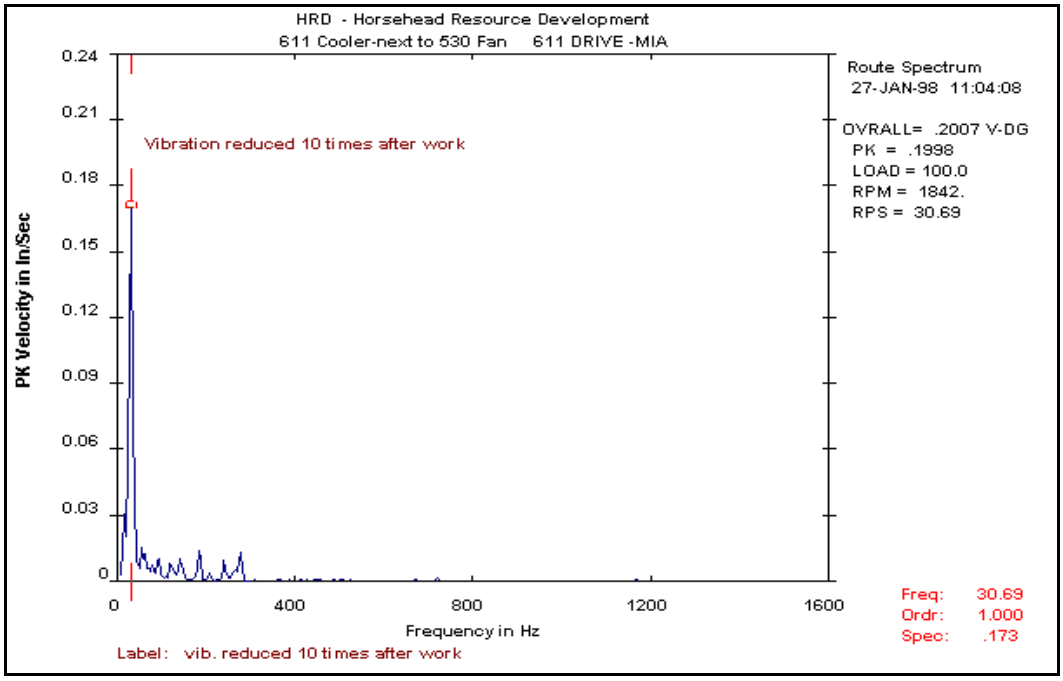
Belt Case 4

It was discovered that both sheaves where worn and needed to be replaced. The sheave was replaced on the Motor, but not on the Gearbox. New belts where installed and straight edge alignment was performed. As displayed below the vibration was reduced 10 times.



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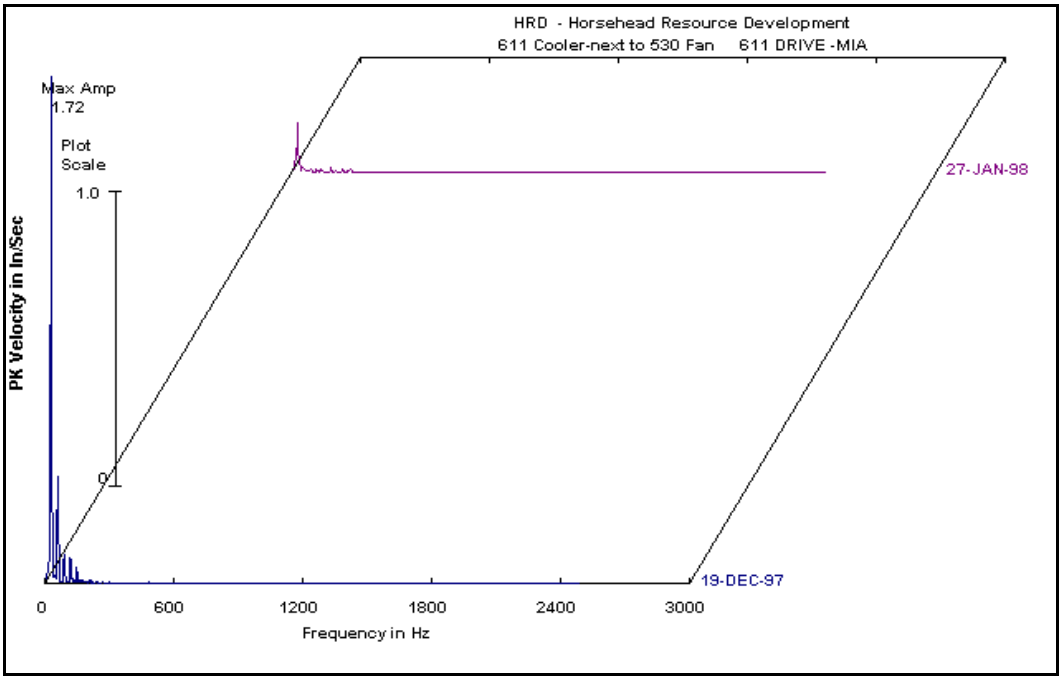
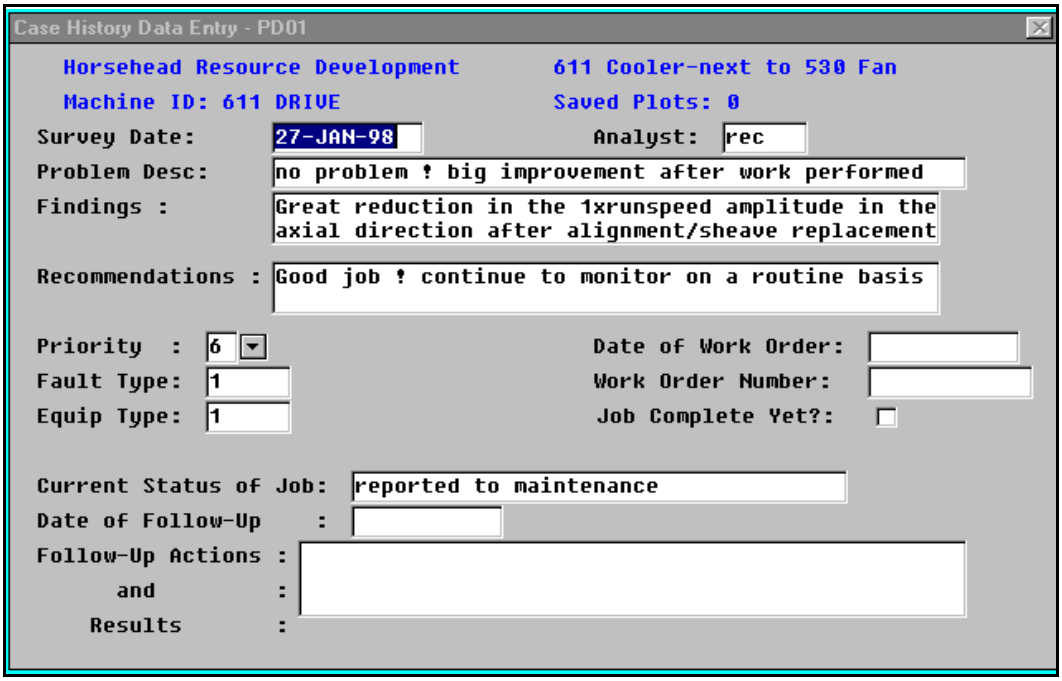
Belt Case 4

This data shows the improvement achieved on the system. The plot at the top is after the repairs were made. The plot on the bottom is before repairs.





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Section Fifteen
Resonance

Objectives

- Define what resonance is and how resonance affects machine vibration.
- Understand the three parameters that affect the amplitude and frequency of the resonance.
- Recognize the single-channel analysis tools available for diagnosing resonance as well as the limitations.

What is Resonance?

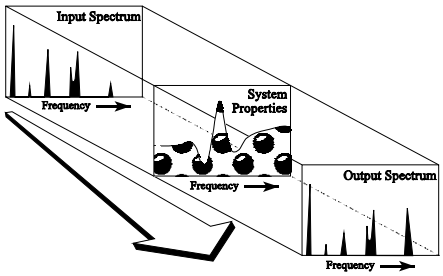
Resonance occurs naturally in all machines, structures, and even animals. Often resonance is called natural frequency or critical frequency.

Resonance is a frequency at which vibration in a structure is amplified.

Several factors work together allowing resonance to occur, such as low stiffness and/or low damping at the resonant frequency. Resonance is not necessarily a problem unless machine defects create vibration or nearby machinery transmits vibration at the same frequency as the resonant frequency.

Resonance does not create vibration; it only amplifies it.

Frequency Response and Transfer Functions



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Resonance is not itself a defect, but it is a property of the whole mechanical system. Therefore, resonance will not be a problem unless some other vibration excites the resonance. This is just like a bell or a drum. The bell possesses the potential to resonate at a particular frequency (or frequencies) if it is excited by being struck. If it is not excited, then it does not vibrate at its resonant frequency.

Resonance is a property of the machine whether running or not. Be aware that the dynamic shaft stiffness when the machine is running may be different enough from the at-rest static stiffness to cause the resonance to vary slightly.

The rule of thumb has always been that a resonant frequency measured with machinery shut off should be at least 20 percent away from any forcing frequency.

Individual parts of a machine have resonant frequencies such as shafts, rotors, casings, and foundations.

When these machines are assembled these resonant frequencies shift because of the mass, stiffness, and damping effects that occur when the machine is put together.

Also, dynamic stiffening effects, as mentioned earlier, may shift the static resonant frequencies when the machines are running at their operational speed. Most machines are designed so that the rotor does not have the same resonant frequencies as the shaft.

Once the two are assembled, the machine is not designed to run at the resonant frequency. However, as machines wear and clearances change, resonances often shift close to operating frequencies.

An unexpected defect frequency, such as a harmonic of looseness or some other machine defect, may excite a resonant frequency.

Machinery vibration at acceptable levels can be amplified to unacceptable levels if the vibration excites a resonant frequency.

For example:

A two-speed machine runs at 900 RPM and 1200 RPM. The machine has a resonance at 1200 CPM which amplifies the 1xTS vibration. At 900 CPM, the 1xTS vibration is 0.1 IPS and 1200 CPM and the resonance causes the 1xTS vibration to increase to 0.5 IPS.

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Resonance may be observed as a machine starts up and runs through a resonant frequency.

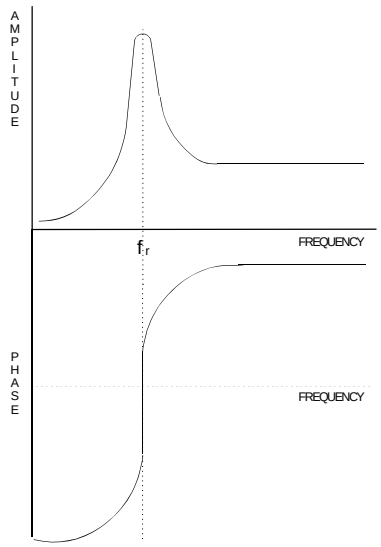
The amplitude of the 1xTS will increase to a maximum amplitude at the resonant frequency and decrease as it passes through the resonant frequency.

The phase will go through a 180° change as the shaft passes through resonance with a 90° phase shift occurring at the resonant frequency.

The 180° phase shift often only occurs on simple single-plane types of rotors.

More complex shaft/rotor systems exhibit a phase shift, although not 180°.

Whenever an analyst is diagnosing a high amplitude vibration problem, the analyst needs to consider the possibility of acceptable vibration exciting a resonance and causing unacceptable levels of vibration.



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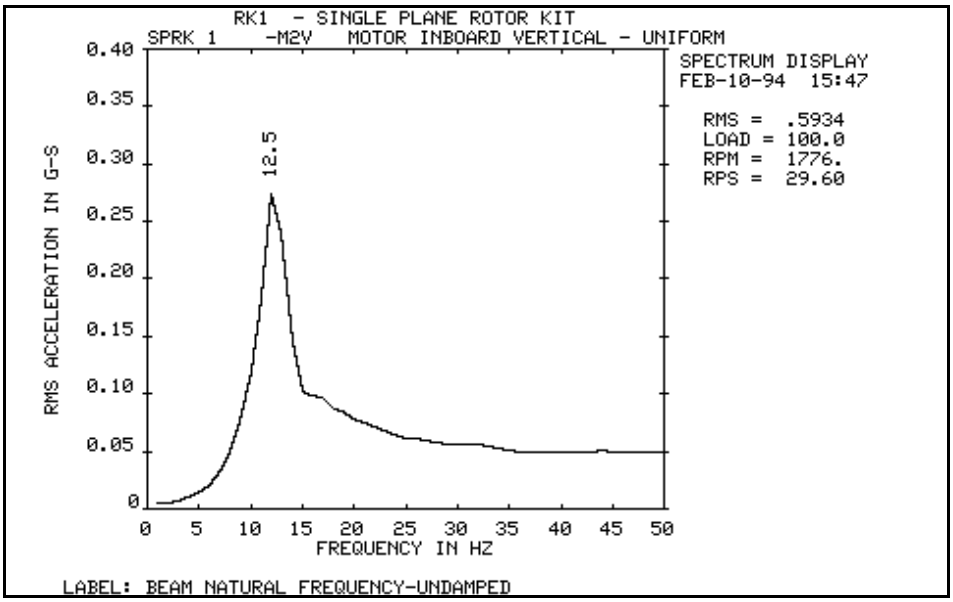
Mass, Stiffness, and Damping

Mass, stiffness, and damping are the three parameters that affect the frequency and amplification of a resonance.

Mass is the property that creates the vibration forces.

Stiffness is the property that counteracts the inertia due to the mass forces.

Damping is the property that converts mechanical energy to thermal energy, actually absorbing the vibration.



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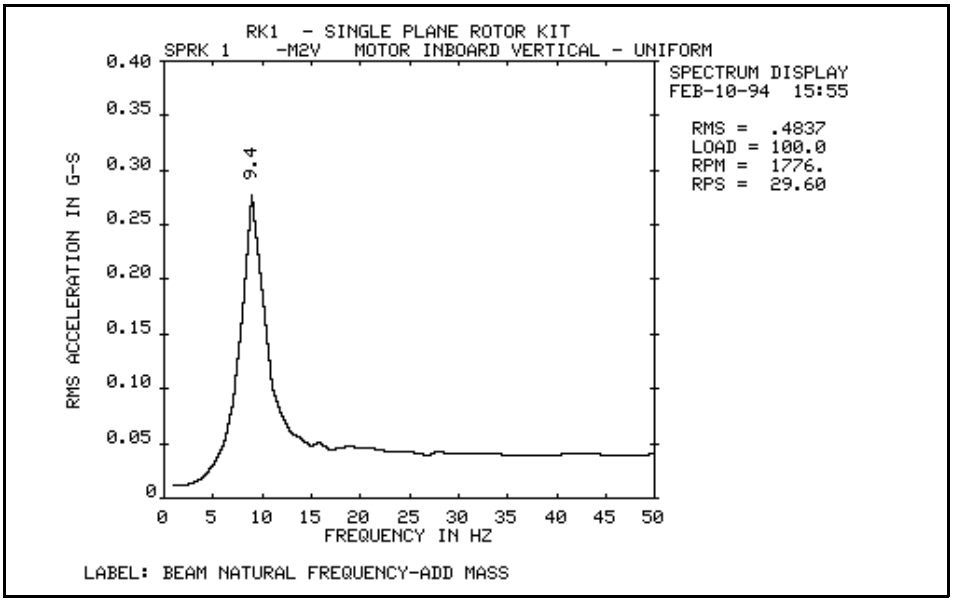
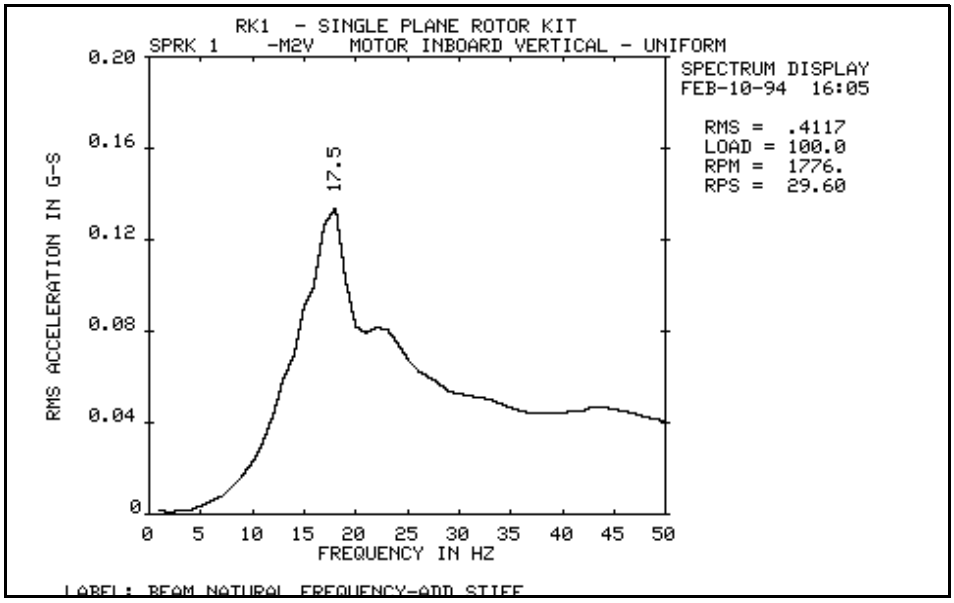
Increasing the mass of a structure will decrease the resonant frequency.



Increasing the stiffness of a structure will increase the resonant frequency.



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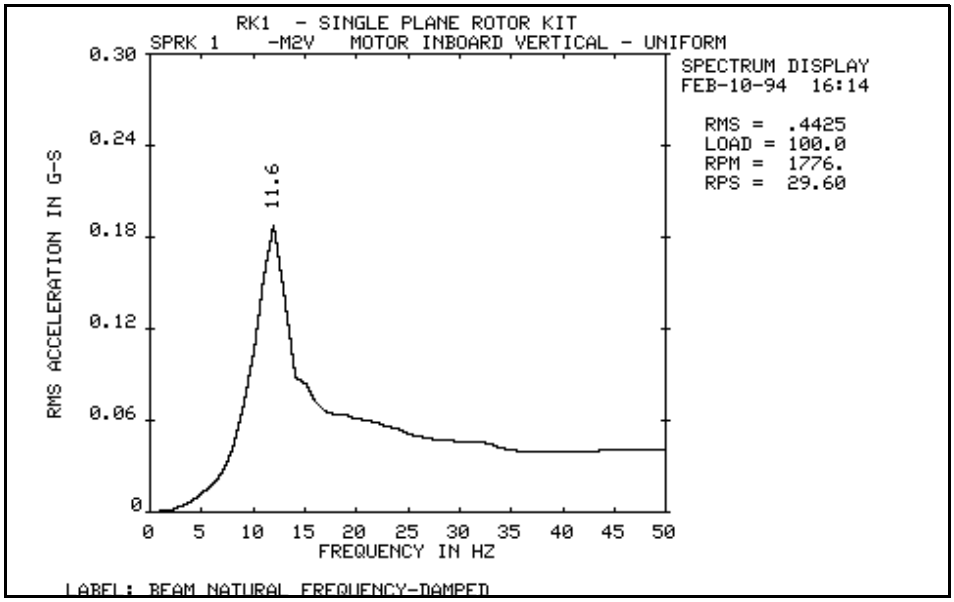
Increasing the amount of damping in a structure will decrease the amplitude of the resonance.

At resonance, damping is the only property that controls the amplitude of the vibration.



The added damping also reduced the resonant frequency slightly.

If adding mass will decrease a resonant frequency, then taking mass away will certainly increase the resonant frequency. Likewise, if adding stiffness increases a resonance then decreasing the stiffness will decrease the resonant frequency.

An analogy could be made to a guitar string.

The tighter the string (the more stiffness) the higher the tone (resonant frequency) that the string will produce when it is plucked. If a thicker string is used (more mass), the lower the tone will be.

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Measuring Resonance

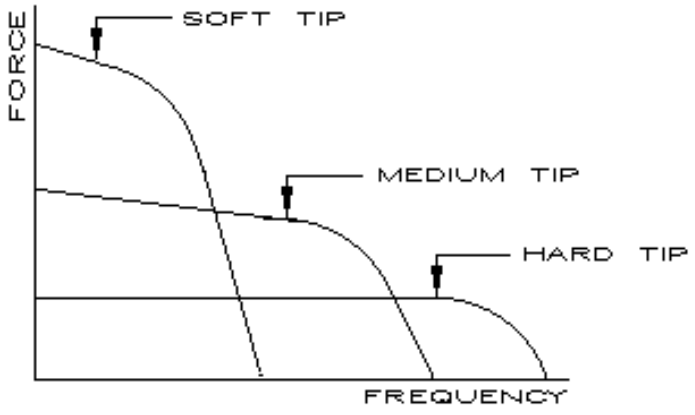
One of the commonly used methods for measuring a structure's resonant frequencies is to impact the structure with a hammer (not a steel hammer) and measure the response.

This works because the impact inputs a small amount of force into the machine over a specific frequency range.

When performing this technique, it is important to try impacting different locations on the structure since all a structure's resonant frequencies will always be measurable by impacting at one location and measuring at the same location. Both drive-point and transfer-point measurements should be tried when attempting to identify machine resonances.

If the hammer has a soft tip, more energy will be input at lower frequencies. Harder tipped hammers place less energy at each frequency, but allow energy to be input at higher frequencies.

Only the response to the impact is being measured with a single-channel analyzer, so the machinery must be shut off to do this properly, unless the data collection is being performed with negative linear averaging.



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Phase is one of the parameters that can clearly identify resonant frequencies.

However, phase cannot be measured during an impact test with a single-channel analyzer; and, in order to confirm the presence of a shaft resonance, the phase must be collected using other methods.

When using a single-channel analyzer, measuring amplitude and phase of 1xTS during a coastdown (or start-up) will identify a resonant frequency as an amplitude increase at the resonant frequency and a phase change of 180° from below resonance to above resonance.

This allows the analyst the two pieces of information needed to identify a shaft resonant frequency. If the phase does not change, a structural resonance and not a shaft resonance is probably being measured.

In either case, it is recommended that a **"**bump test**"** be performed in addition to the startup/coastdown data.

Resonance is most definitively measured using a multichannel analyzer to measure impact and response data at the same time, thus allowing for phase and coherence data to be collected at the same time.

Coherence is a dual-channel function that relates how much of the input signal caused the output signal. This means that resonance frequencies can be identified more rapidly.

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Machinery Considerations

Some consideration needs to be given to different types of machinery and some conditions that may make resonant testing difficult. Due to structural differences between the horizontal and the vertical directions, the resonant frequencies may vary slightly from the horizontal to the vertical direction. So resonances can be directional.

As discussed earlier, resonant frequencies may vary slightly as machinery is running or stopped. Vertical equipment is generally very troublesome because the equipment, being vertical mounted, has a resonance resulting from the cantilever action of the motor.

Some machines are too massive to excite with a hammer and require alternative means to excite the resonant frequencies. Sometimes on very large structures using a dynamic shaker setup to vibrate through a frequency range is used because shakers are able to place more energy in at each frequency.

One last consideration — before performing resonance testing it is very helpful to measure the background vibration levels. This can prevent misdiagnosing a large amplitude frequency in the background data as a resonant frequency that may need to be corrected.

Summary

This section has discussed the effect of resonant frequencies on machinery vibration. All structures have resonant frequencies, but the resonances will not affect a machine unless there are frequencies present that excite the resonances. If machine vibration excites a machine's natural frequency, three options exist.

❶ Move the forcing frequency away from the resonant frequency.

❷ Move the resonant frequency away from the forcing frequency.

❸ Add damping to reduce the amplification factor of the resonance.

Options 2 and 3 generally involve some structural design changes that should not be made unless a modal analysis and/or a finite element analysis study has been performed on the structure.

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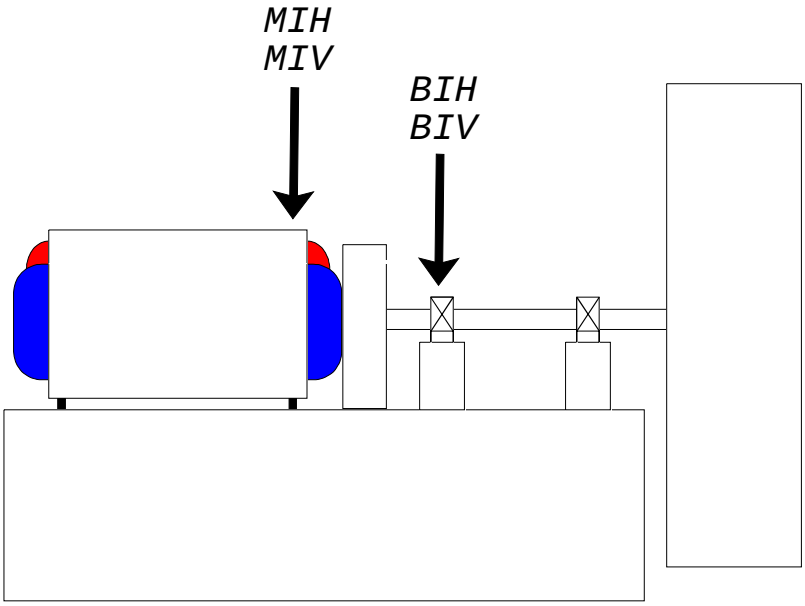
Case Histories - Resonance

Process Vacuum Blower Resonance Case #1



❶ 125 HP, 1200 RPM motor.

❷ Unit a direct-driven overhung fan.

❸ Directional vibration was present.

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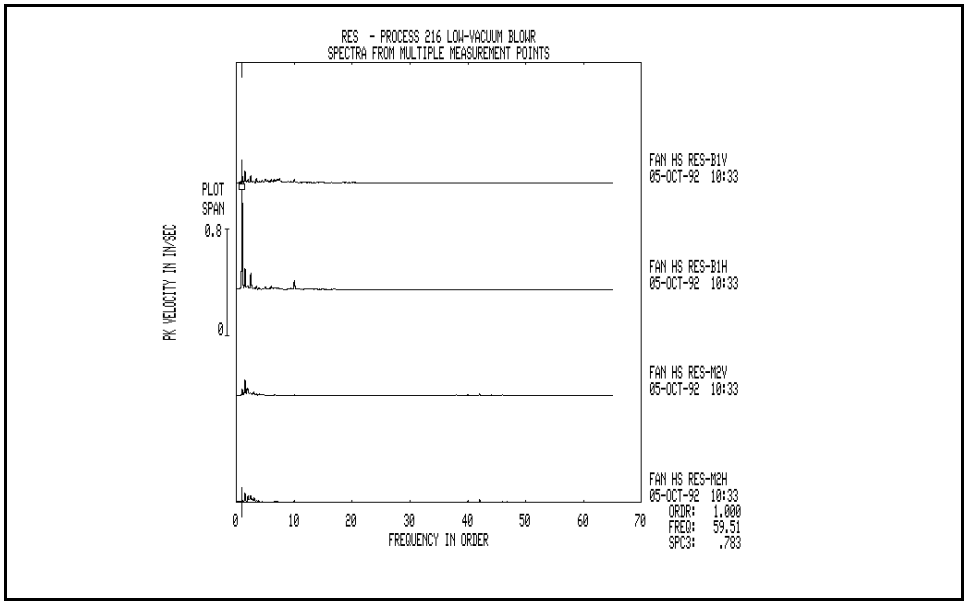
Resonance Case #1



The above multiple-point plot shows the spectra from the motor and fan inboard points. Notice the large difference in the horizontal and vertical 1xTS amplitudes.

The points most affected are the points on the fan.

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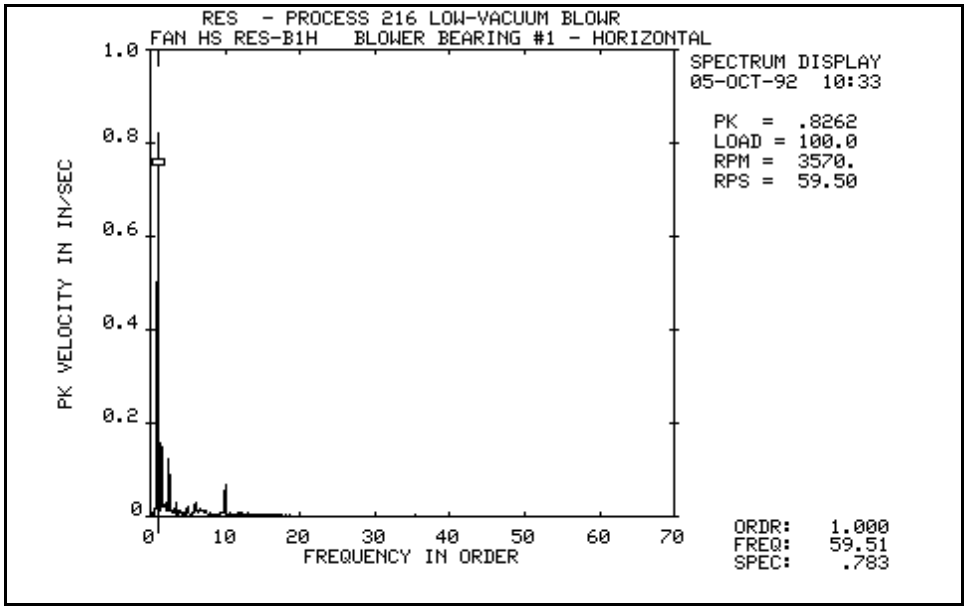
Resonance Case #1



The full-screen spectrum from the blower inboard horizontal is displayed.

The cursor shows the 1xTS amplitude to be over 0.7 IPS. That is approximately ten times the vertical amplitude.

The next page shows the results of the impact test performed on the blower housing.

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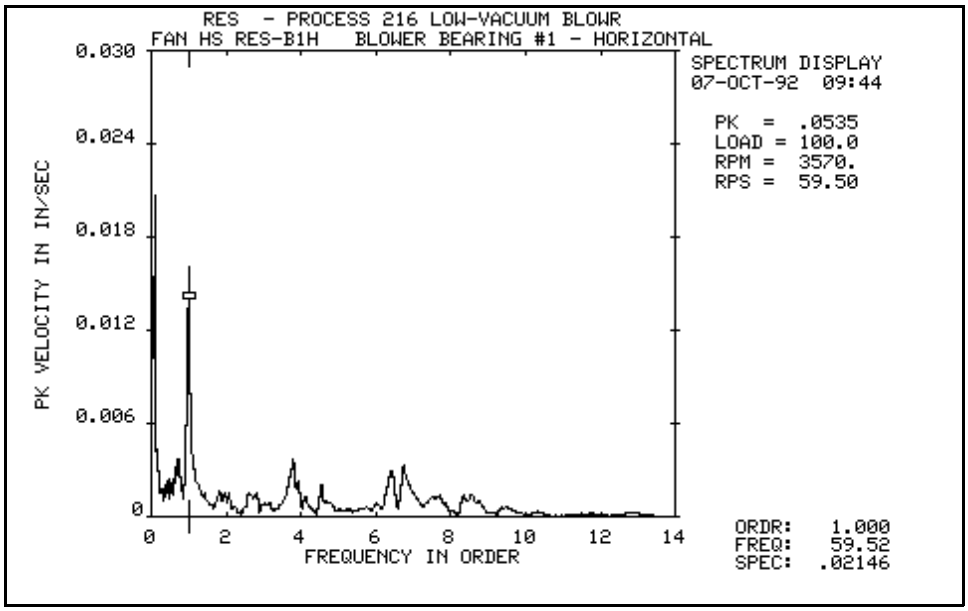
Resonance Case #1



The impact spectrum clearly shows the highest peak is at motor turning speed.

This resonance of the housing amplifies the turning speed vibration.

The waveform for this impact is displayed on the next page.

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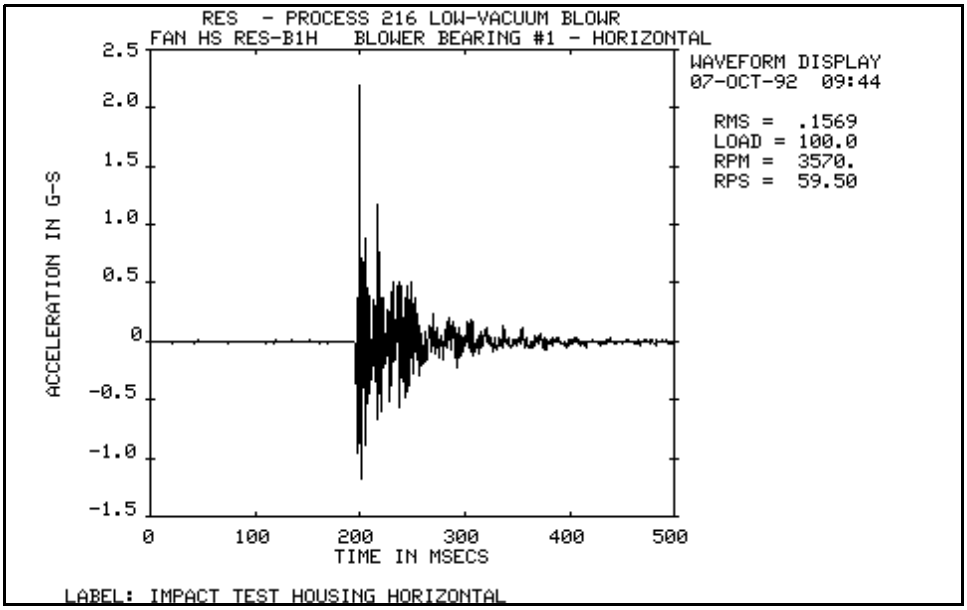
Resonance Case #1



The display above shows the waveform for the housing impact. The flat line before the data represents the pre-trigger setup in the analyzer.

Normally set to 35% pre-trigger.

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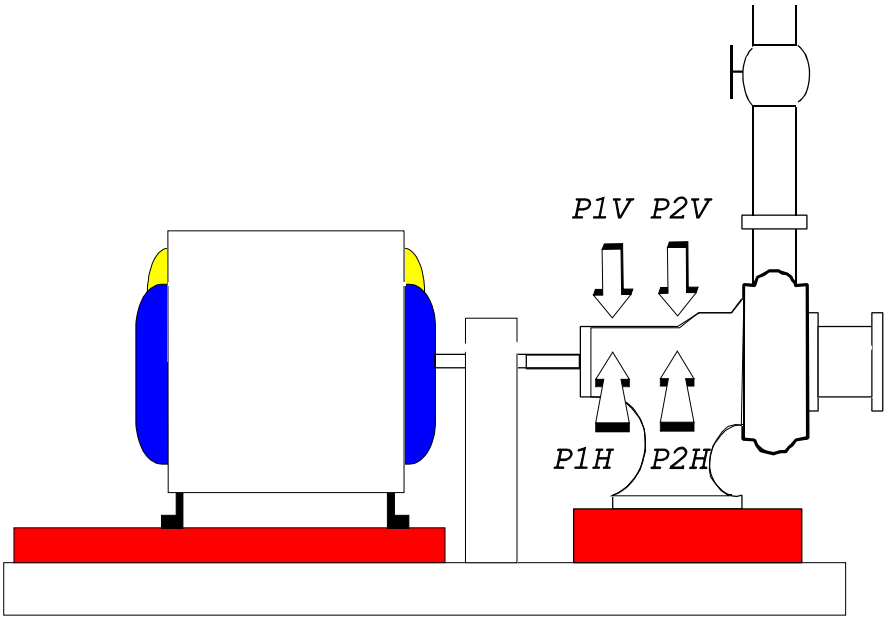
Process Pressure Pump Resonance Case #2



❶ 20-HP, 3600 RPM motor.

❷ Unit is a direct-driven overhung pump.

❸ High directional vibration.

❹ The pump impeller has three blades.

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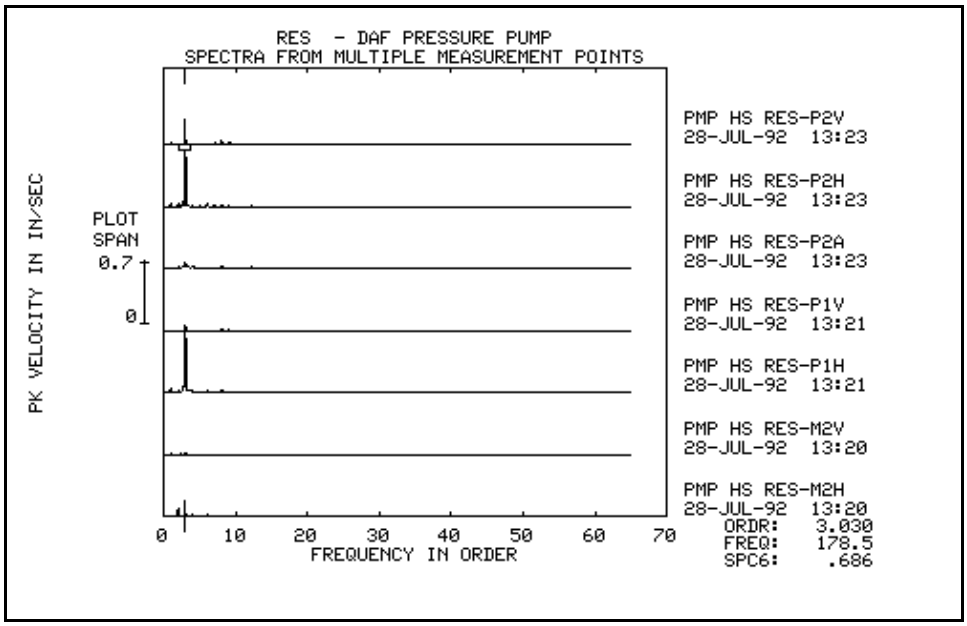
Resonance Case #2



The multiple-point plot above shows the motor and pump data.

The highest vibration is on the pump horizontal points.

The cursor shows the peak to be located at 3xTS.

A multiple plot showing just the horizontal pump points is on the next page.

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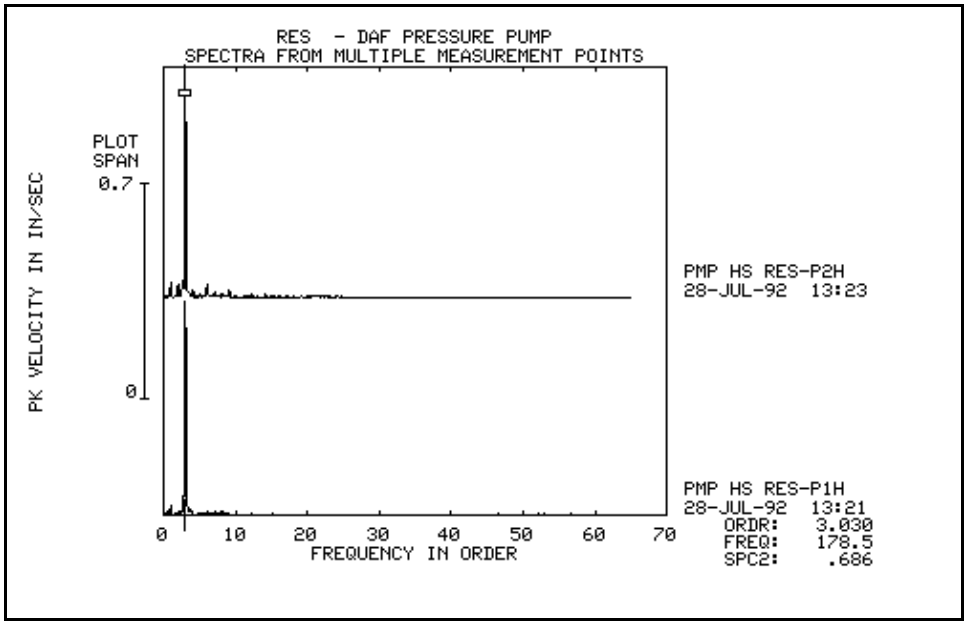
Resonance Case #2



The pump horizontal data is displayed in the above plot.

The cursor marks the 3xTS peak on the pump outboard horizontal.

The large differences between the horizontal and vertical readings suggest a possible resonance problem.

Remember the Rule, when there a ratio of 3 to 1 or greater when comparing the Horizontal reading to the Vertical reading, suspect resonance.

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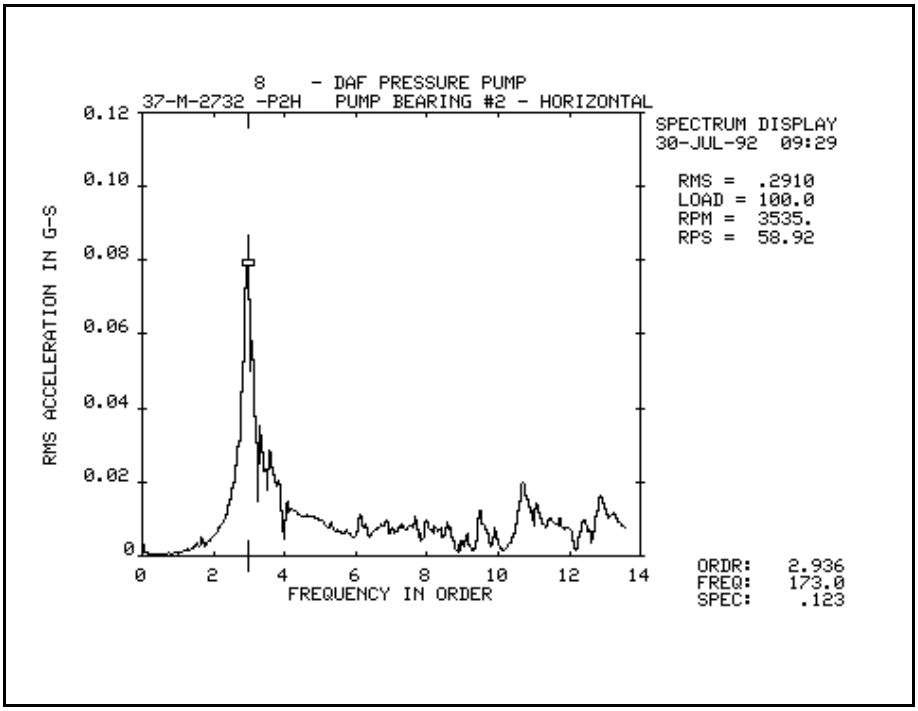
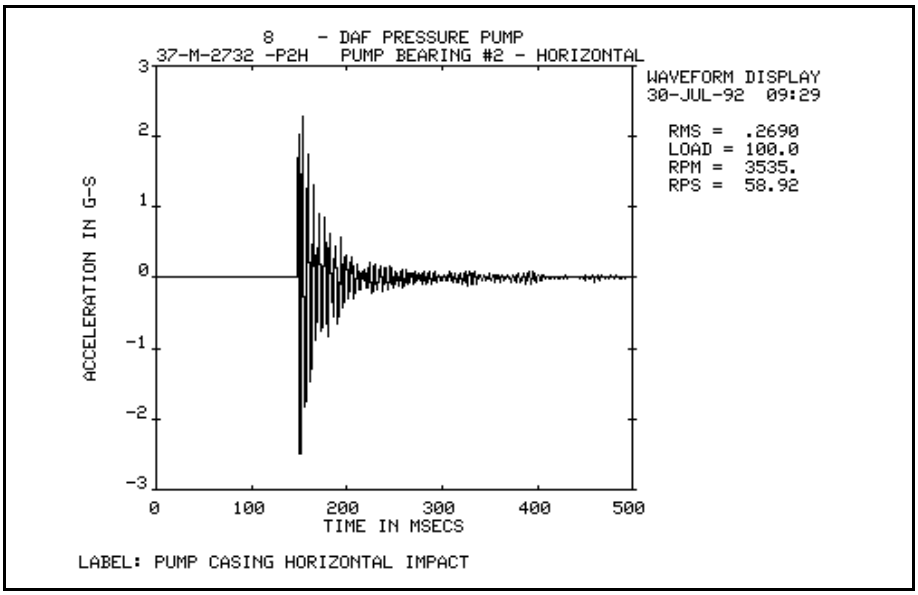
Resonance Case #2





The spectral and waveform data are presented above. The analyst used an accelerometer to collect the impact response. Notice the spectrum shows the resonance peak more clearly in units of acceleration.

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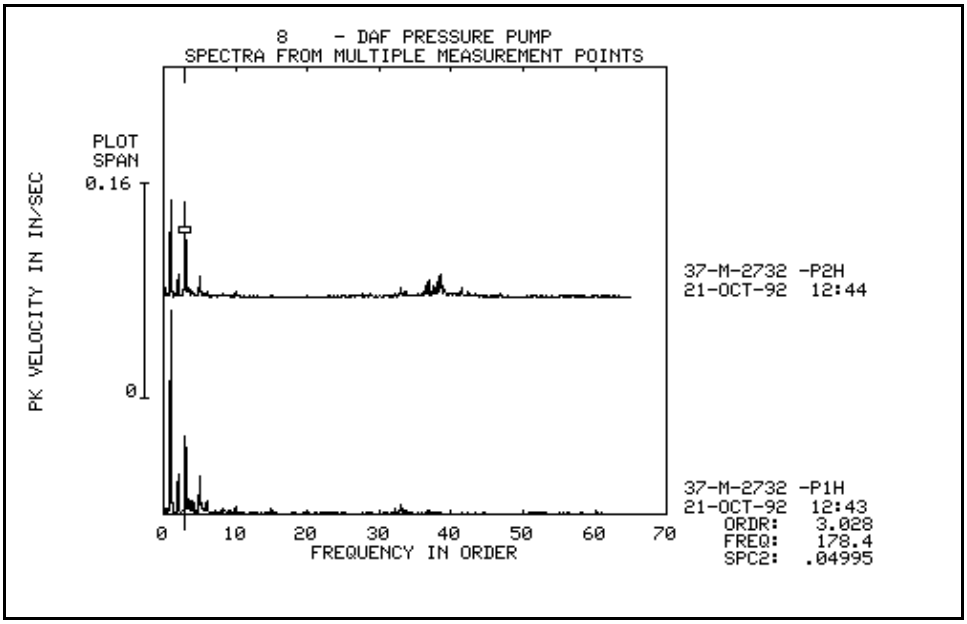
Resonance Case #2



The data above shows the after repair vibration on the horizontal pump points.

The three-bladed impeller was replaced with an impeller that had five blades.

This moved the forcing frequency caused by the pump's blade pass to above the pump casings resonant frequency.

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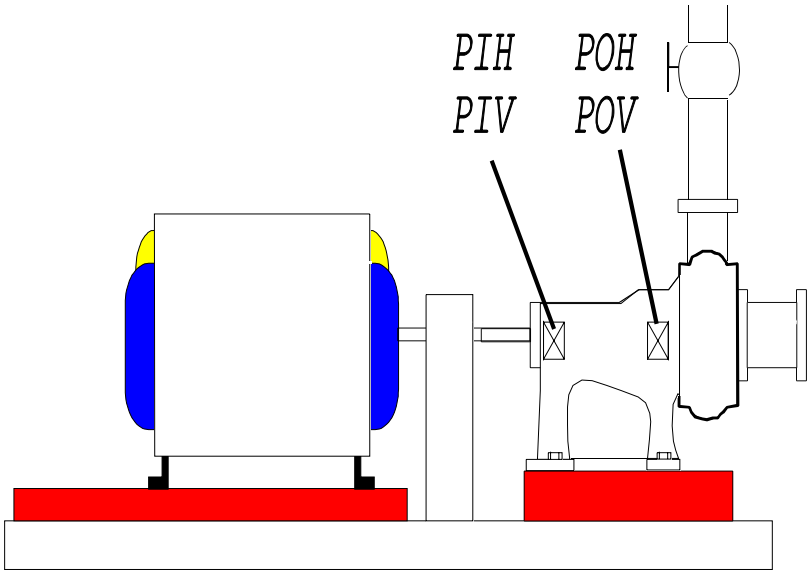
Reboiler Feed Pump Resonance Case #3



❶ 50 HP, 3600 RPM motor.

❷ Unit is a direct-driven overhung pump.

❸ High directional vibration is present.

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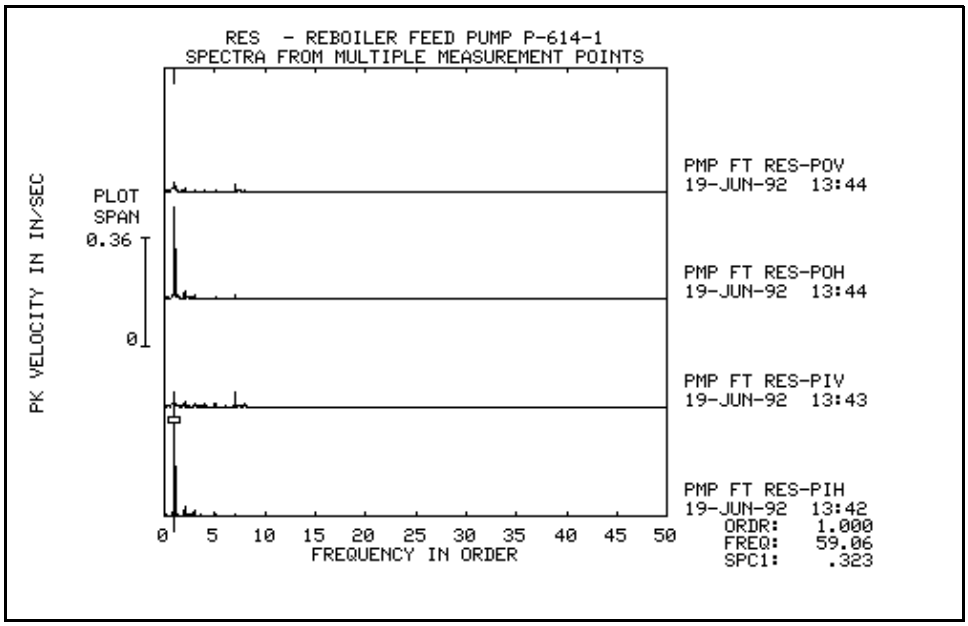
Resonance Case #3



The multiple-point plot from the pump shows the vibration is primarily at 1xTS and in the horizontal direction.

The next page displays the Pump Inboard Horizontal (PIH) measurement point.

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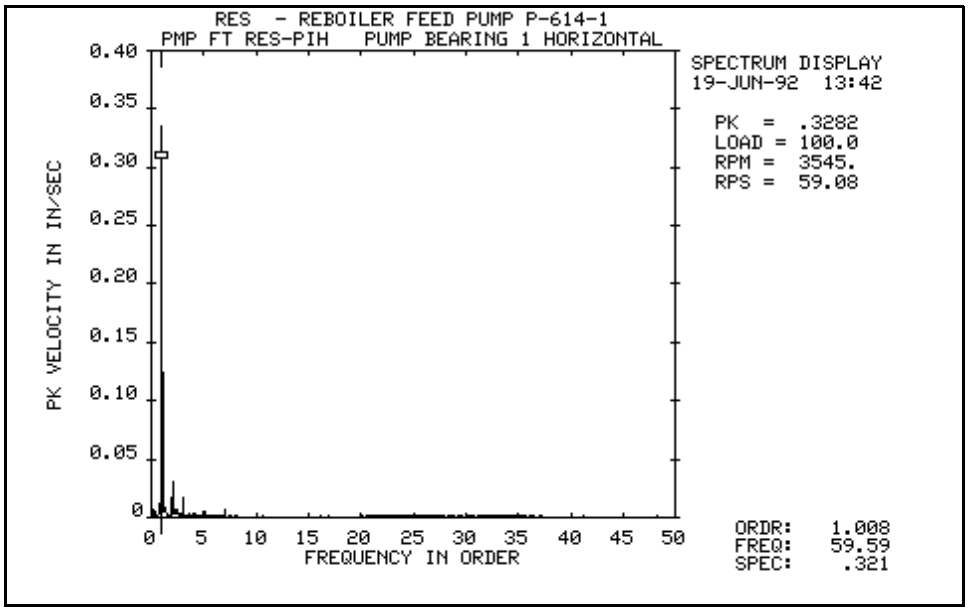
Resonance Case #3



The cursor marks the amplitude of the 1xTS peak at .31 IPS.

This spectrum has all the characteristics of a balance problem.

If this was unbalance, the amplitude of the 1xTS peaks in the vertical direction should be higher.

The directional vibration is caused by resonance, but the excitation force is different.

This resonance was caused by improper mounting of the pump. In this case, one of the pump foot bolts was stressing the casing because its hole was drilled at an angle in the base.

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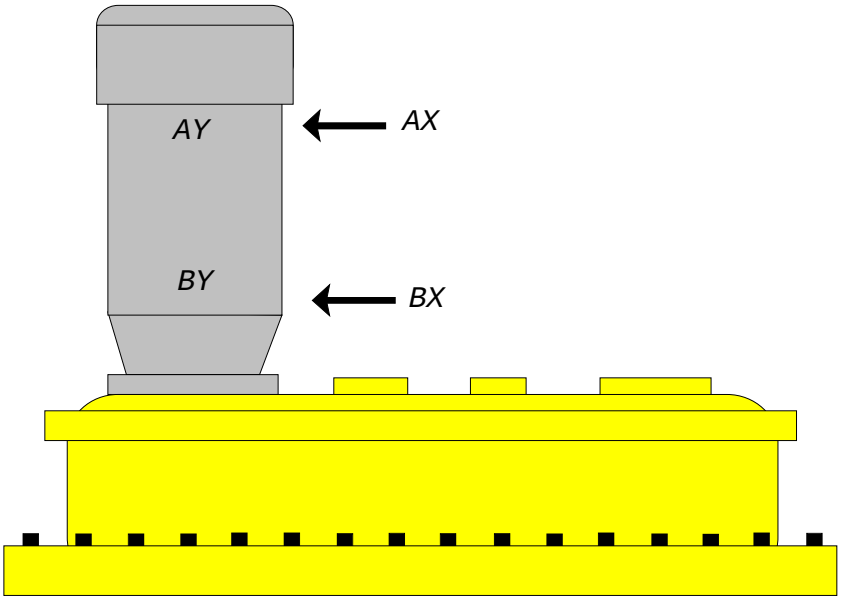
Aerator
Resonance Case #4



❶ 250 HP, 1200 RPM motor.

❷ Gearbox has two intermediate shafts.

❸ High directional vibration is present.

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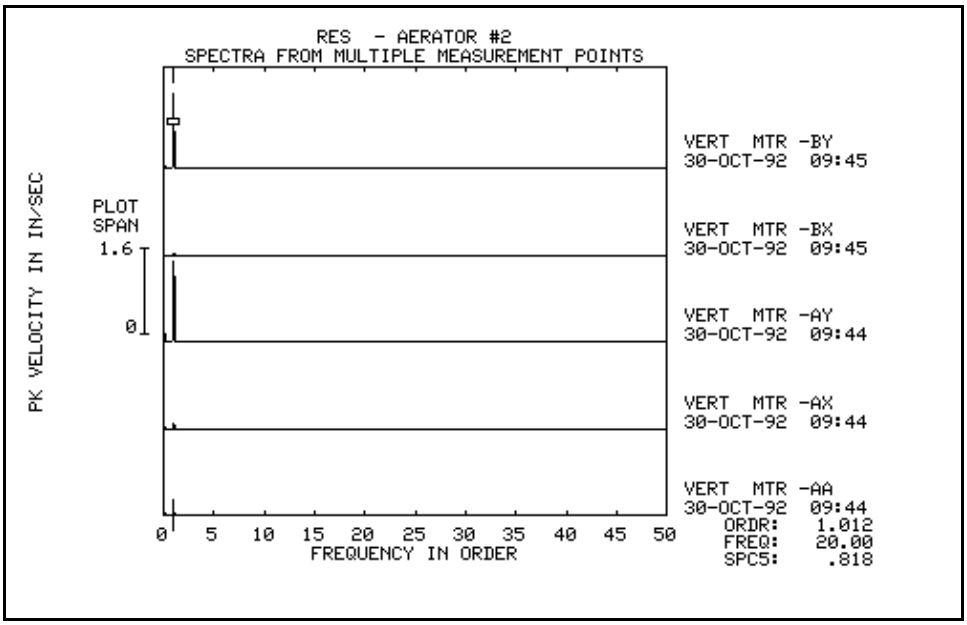
Resonance Case #4



A multiple-point spectrum shows the presence of high vibration at 1xTS in the radial Y direction.

The vibration at shaft turning speed is not present in the axial measurement.

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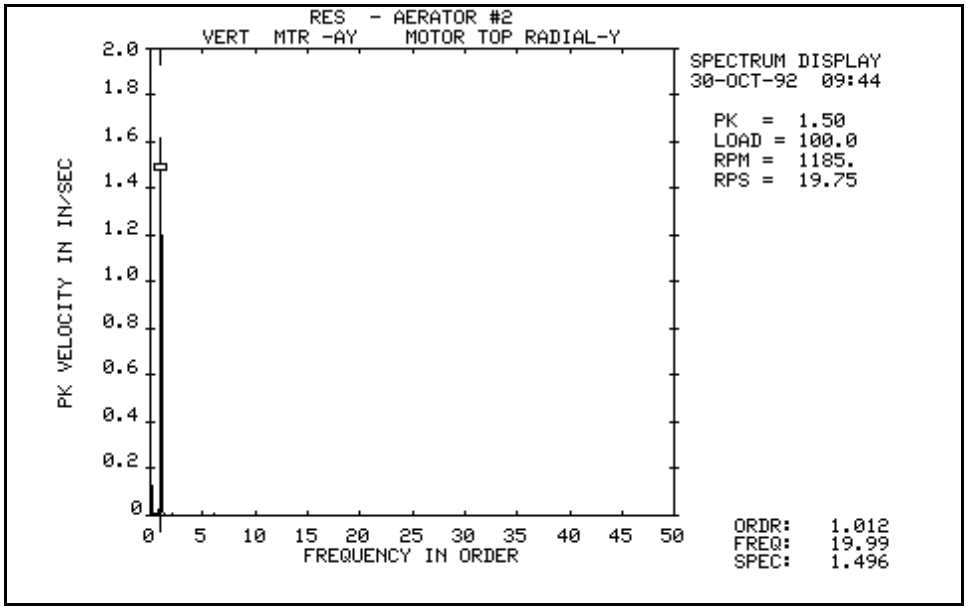
Resonance Case #4



A full-screen spectrum of the upper motor bearing's radial Y position shows the 1xTS peak is almost 1.5 IPS.

This directional flexibility is not uncommon in vertical equipment due to their increased sensitivity to resonance.

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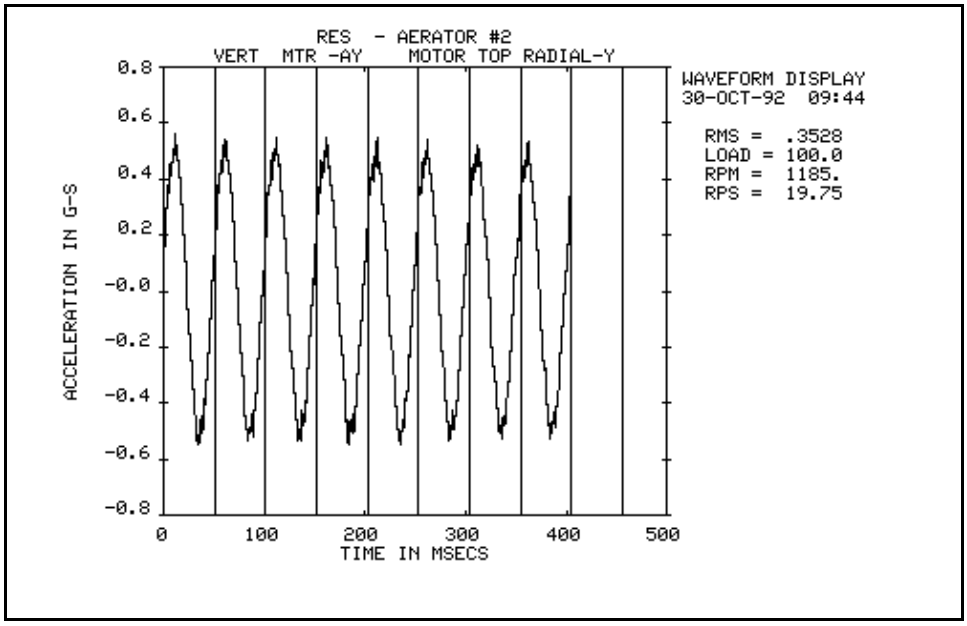
Resonance Case #4



The waveform shows a repeatable, sinusoidal pattern that lacks complexity. The amplitude is relatively low and is free of impacts.

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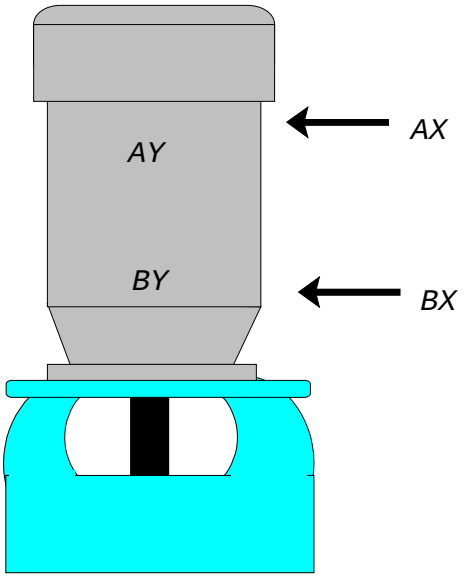
Effluent Pump
Resonance Case #5



❶ 60 HP, 3600 RPM, motor.

❷ Vertically mounted well pump.

❸ Directional vibration is present.

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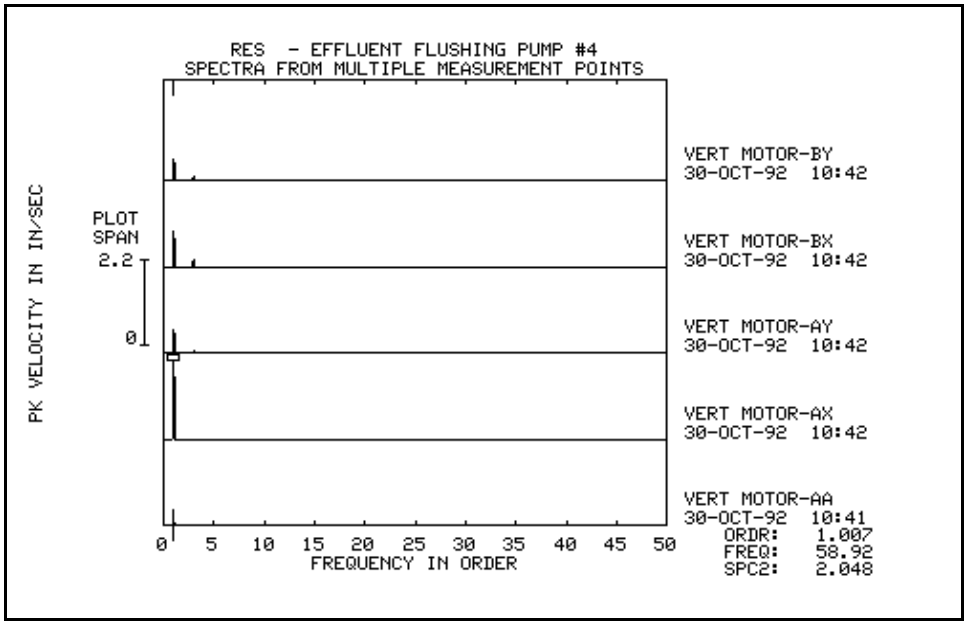
Resonance Case #5



The multiple-point plot for the effluent pump is displayed above. The cursor marks the running speed peak at 1 order.

The amplitude of this peak is over 2 IPS.

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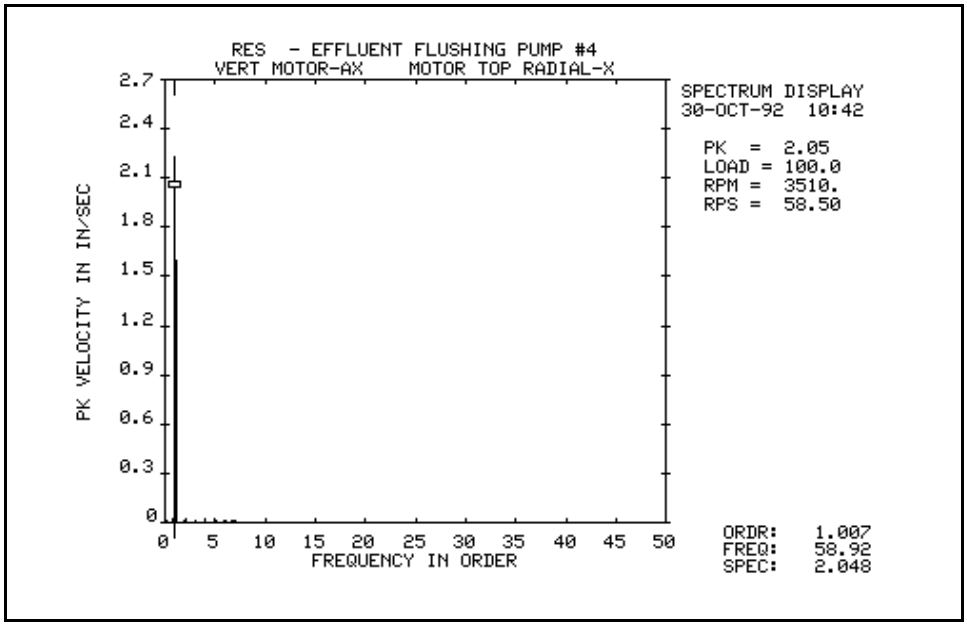
Resonance Case #5



The full-screen spectrum for the upper motor bearing point AX is displayed.

The overall vibration level for the plot is 2.05 IPS and the 1xTS peak comprises almost all this value.

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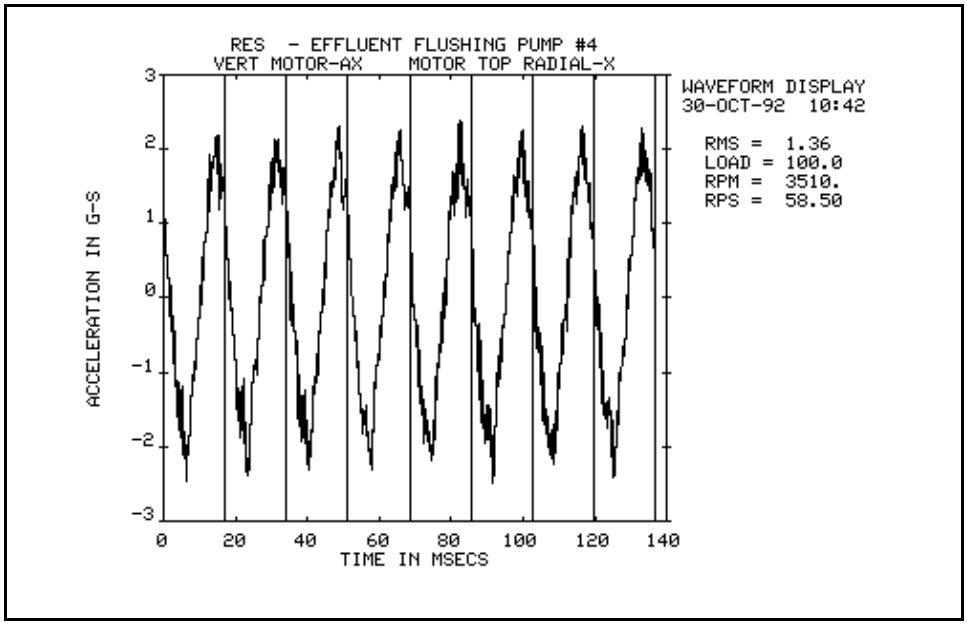
Resonance Case #5



The waveform shows the sinusoidal pattern with the one distinct peak per shaft revolution.

This repetitive shape, however, lacks any real complexity.

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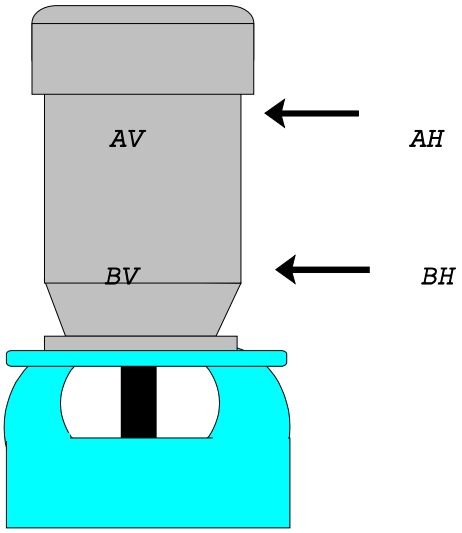
500 HP Vertical Pump
Resonance Case #6



❶ 500-HP, 1200 RPM motor.

❷ Motor was just refurbished.

❸ Data was taken for certification.

❹ High directional vibration was present.

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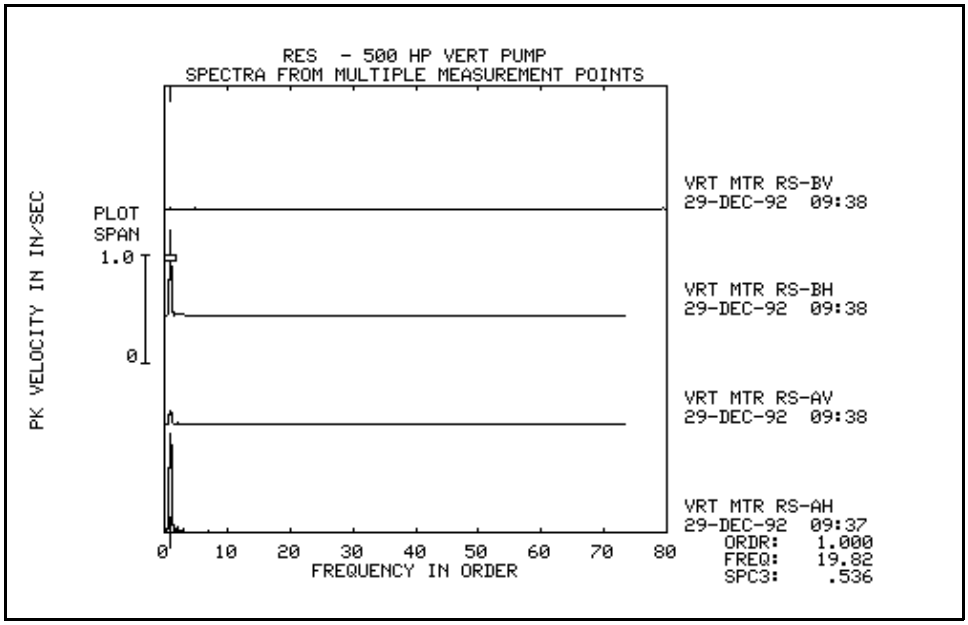
Resonance Case #6



The certification data for the motor is displayed on the multiple point plot.

The cursor marks the 1xTS peak on the measurement point BH.

Again, the high vibration is only present one direction.

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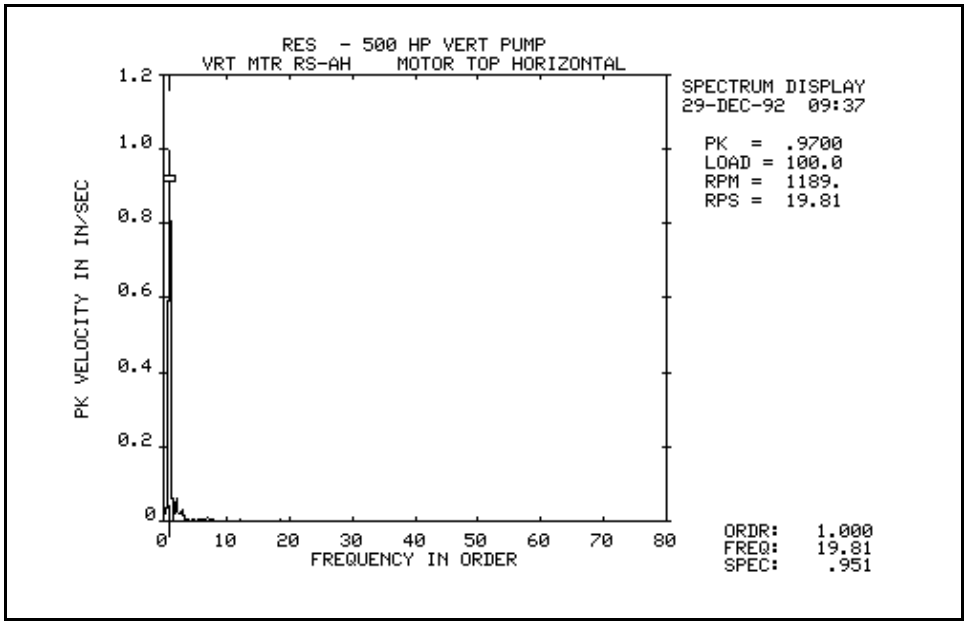
Resonance Case #6



The data for the upper motor bearing is shown with a cursor marking the 1xTS peak.

The analyst inquired about the support structure the unit was mounted on. He was told the pump was mounted on an I-beam support that could move in the direction of the vibration.

Additional support in the horizontal direction was recommended to reduce the vibration.

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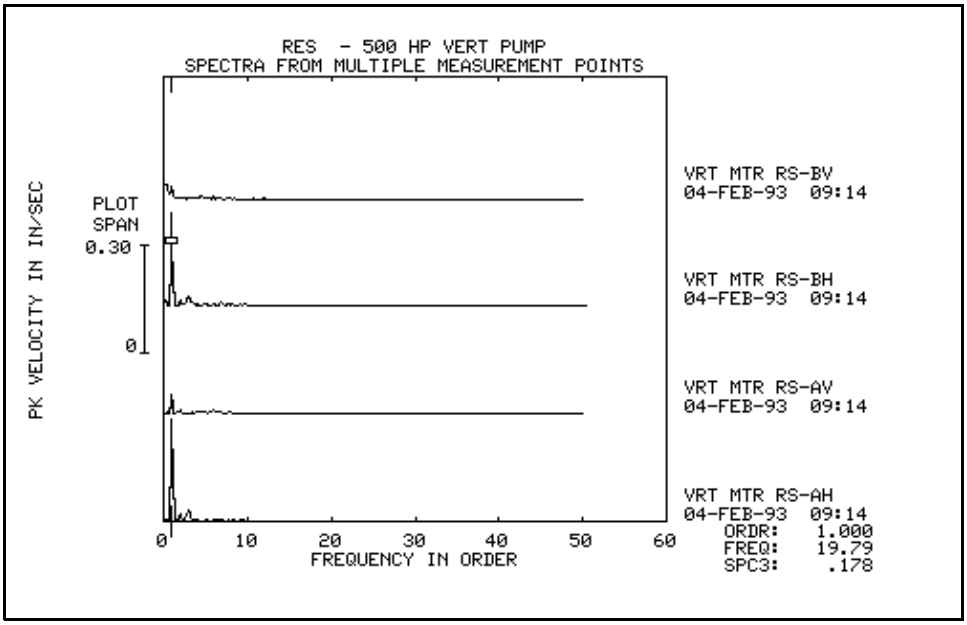
Resonance Case #6



This multiple-point plot for the motor was collected after temporary supports were installed.

The cursor marks the 1xTS peak for the lower motor bearing's measurement point. Remember the initial vibration level was 0.5 IPS.

The supports reduced the vibration by a factor of three.

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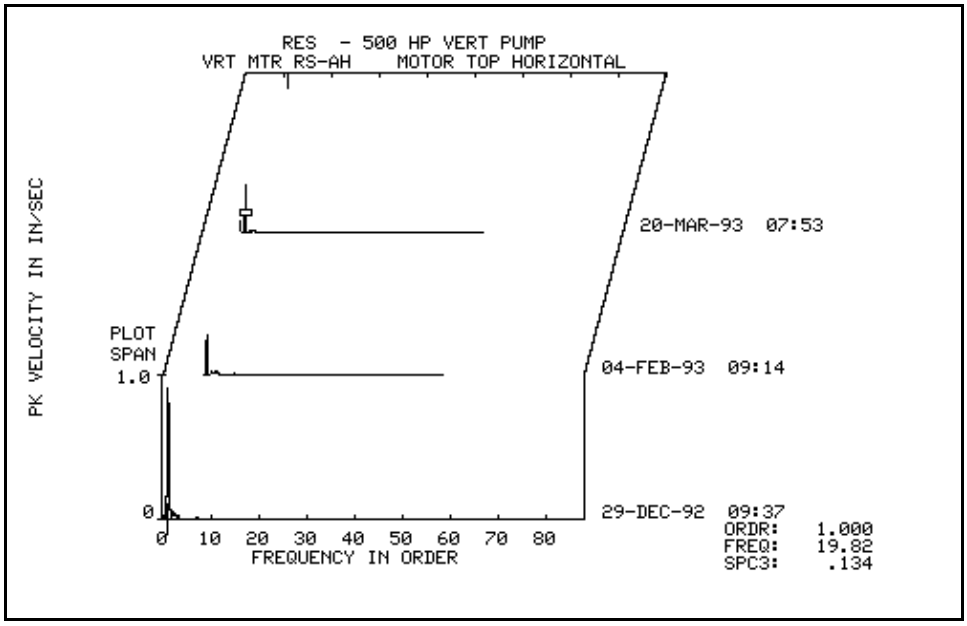
Resonance Case #6



The multiple-spectrum plot for measurement point AH is displayed showing the three readings that were taken.

December's data shows the unit after the motor was repaired.

The amplitude of the 1xTS peak was .9 IPS.

The data for March shows the same measurement point after the supports were permanently installed.

The amplitude of the 1xTS was reduced by a factor of seven to a value of 0.13 IPS.

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Section Sixteen
Pump Vibration Basics

Objectives

- Describe the primary vibration generation principles that cause vibration in centrifugal pumps.
- List vibration characteristics that point to pump defects.

Fundamentals of Centrifugal Pumps

Centrifugal pumps are commonly used in industry for fluid movement. Pumps can exhibit typical mechanical vibration generated by defective bearings, mechanical unbalance, looseness, etc. This section is dedicated to vibration related to the hydraulic forces occurring within the pump and the piping system.

Vibration is caused by the interaction between the moving pump impeller and the stationary parts of the pump such as the volute and the diffuser vanes. Also, vibration is caused by the interaction between the impeller blades and the fluid being pumped.

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Pump Operation

In a centrifugal pump, a rotating element called an impeller is enclosed in a case. The fluid to be pumped enters into the case through the suction piping into the impeller and is forced out the discharge side of the pump by means of centrifugal force pushing the fluid out radially through the impeller. Several components within the pump can generate vibration. Some of the vibration is broadband and some is discrete.

Impeller Basics

Single-stage pumps use one impeller to create the necessary head and flow. Pumps with more than one impeller are called multistage pumps. Multiple impellers are used when the system requires more head than one stage can provide.

Different types of impellers are used, depending on the type of flow required.

❶ Radial Flow Impellers - the fluid enters the impeller radially.

❷ Axial Flow Impellers - the fluid enters the impeller axially (through the eye of the impeller).

❸ Mixed Flow Impellers - a combination of Radial and Axial Flow.

❹ Single Suction Impellers - the fluid enters the impeller from only one side.

❺ Double Suction Impellers - the fluid enters the impeller from both sides.

Other characteristics affect the impeller design such as the enclosure of the impeller and the orientation of the shaft (vertical or horizontal).

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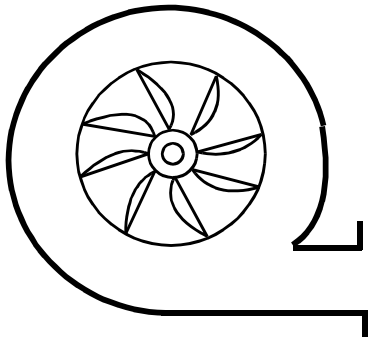
Pump Casing

The pump casing forms the pressure boundary around the fluid. The pumped fluid needs to have some of its velocity energy converted to pressure energy. Typically, this is done through the use of a volute or diffuser vanes.

The casing can also be described by the way it must be separated for impeller access— radial or axial split casing.

Volute Design

A volute is an ever widening spiral casing surrounding the impeller that reaches its greatest area at the pump discharge. The increase in area converts some of the velocity energy to pressure energy. Some pumps have twin volutes that reduce excessive flow turbulence caused when the pump operates away from its best efficiency point (BEP).



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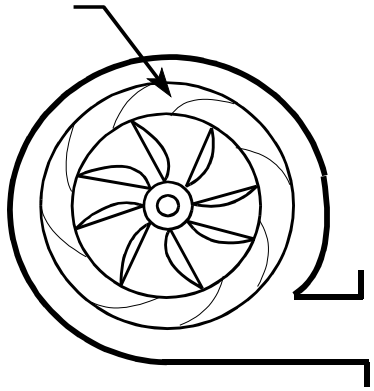
Diffuser Vanes

These are stationary vanes at the discharge of the impeller that convert some of the velocity energy into pressure energy. No volute casing is needed when the diffuser vanes are used.

Diffuser vanes work well only if the pump is operating near the design point so that the angle that the fluid is leaving the impeller vanes is the same angle that it is entering the diffuser vanes. If the pump is not operating within 5 to 10 percent of its BEP, then the flow path can become turbulent.



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Net Positive Suction Head (NPSH)

NPSH is a measure of the difference between the total suction head and the fluid vapor pressure. A fluid will flash to vapor if the pressure that the fluid is under drops below the vapor pressure.

If the suction pressure of the pump does not exceed the vapor pressure of the fluid, then fluid will change to a vapor and the pump will cavitate. Pump cavitation causes excessive vibration and noise and will eventually destroy the pump impeller and cause the pump to fail.

There is a difference between the available NPSH and the required NPSH:

❶ ***Available NPSH*** is the difference between the absolute suction head and the vapor pressure of the fluid at its operating temperature. This is a characteristic of the pumping system.

❷ ***Required NPSH*** is the smallest margin necessary between the pump suction head and the vapor pressure. This is a characteristic of the pump design.

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Pump Vibration Analysis

Vane Pass Frequency

When most people think of pump vibration, they think of vane pass as one of the pump's primary defect frequencies. Vane pass frequency is not always a defect frequency, but actually is an operational frequency. It is normal to see this frequency in pump vibration data.

The amplitude of this frequency may increase if the pump is operating away from the BEP. Also, higher amplitude vibration may be present if there is not an equal gap between the rotating impeller and the pump's stationary parts, such as the diffuser vanes.

There are two different vane pass frequencies depending on the pump design.

❶ Impeller Vane Pass Freq. = (# Impeller Vanes) * (Shaft Turning Speed)

❷ Diffuser Vane Pass Freq. = (# Diffuser Vanes) * (Shaft Turning Speed)

The diffuser vane pass is only possible if the pump has diffuser vanes.

Cavitation

Cavitation occurs when the pressure of the fluid drops below the vapor pressure for the temperature of the fluid. When this pressure drop occurs, whether it is a system pressure drop or a localized pressure drop, voids or cavities (bubbles) will form in the liquid. These bubbles implode or collapse when the fluid moves through impeller to the high pressure side of the pump, causing the impeller to erode. This can be very damaging and eventually the impeller will fail.

Vibration due to cavitation is typically seen as harmonics of turning speed up to the impeller vane pass frequency. Higher frequency broadband noise can also be measured as well as heard by the analyst. A steady crackling sound (often compared to popcorn or gravel moving through the impeller) is present.

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The broadband vibration is usually present above the impeller blade pass frequency. Since cavitation is caused by reduced NPSH, the solution is to raise the NPSH above the fluid vapor pressure.

Recirculation

This is a pump problem with symptoms similar to cavitation. However, the problem is very different. Recirculation occurs when the pump has such a high suction pressure that it actually pulls the fluid from the output back through the pump. The fluid can flow backward through seals and even the pump impeller clearance.

The major contributor to recirculation is reduced flow in addition to high suction pressure. The solution to this is usually to increase the fluid flow through the pump or to install a bypass.

Recirculation is a hydraulic source of 1xTS vibration as well as subsynchronous broadband noise and it can produce a random crackling noise similar to what is heard with cavitation. The difference to remember is that, with recirculation, the crackling noise will be intermittent and near the pump inlet while with cavitation, the crackling noise will be noticeable on both the inlet and outlet and will be steady.

Hydraulic Unbalance

Uneven flow distribution entering the impeller can introduce a hydraulic unbalance that will show up at 1xTS in an axial direction. This can be caused by the poor design of suction piping such as an elbow close to the pump inlet. This causes the fluid to enter the pump impeller at different flow velocities. A straight run of pipe equal to between 20 and 40 diameters before the pump inlet is recommended to let the flow equalize before it enters the pump.

Hydraulic unbalance may also occur if the impeller is mechanically balanced but not symmetrical in the fluid flow areas. This is similar to an eccentric rotor defect.

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Flow Turbulence

Flow turbulence may be more difficult to detect. This turbulence is hydraulically induced by low suction pressure due to system problems that produce inadequate NPSH. This generally shows up in a low-frequency region between 0.1 and 0.4 orders of turning speed. Excessive turbulence has been known to produce higher frequency broadband vibration as well.

Turbulence often occurs in pumps with diffuser vanes or a twin volute. If the pump is not operating near its BEP, then the fluid leaving the impeller will not enter the diffuser vanes at the correct angle which in turn causes the turbulence. This tends to lead to higher axial vibration at the subsynchronous frequencies.

Resonance

Resonance is a problem that can occur in pumps as well as piping. Vertical pumps produce classic resonance problems due to the long and slender nature of the structure as well as the low natural frequency of most vertical pump foundations.

Piping can also produce a resonance problem. This can be both a structural pipe resonance and acoustic fluid column resonance. Both of these resonances can be excited by various pump operational and pump defect frequencies.

Structural resonance problems should be corrected with modifications made to the pipe or the pump foundation unless the forcing frequency can be altered to prevent resonant frequency excitation.

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Summary

This section has reviewed some basic pump vibration concepts. The basic components of a pump—the impeller, volute and diffuser vanes—were covered as well as the concept of net positive suction head and its importance to pump vibration.

Vibration frequencies to watch for include impeller vane pass frequency and diffuser vane pass frequency. These frequencies are generally operational and do not indicate pump defects. However, if the amplitude increases dramatically, this can be an indication of a developing problem.

Cavitation and recirculation can be similar, but cavitation generates broadband noise around vane pass frequency and recirculation creates low-frequency vibration.

Resonance can be a problem in pumps just as it can in other structures. Vertical pumps are often prone to resonance problems and structural modifications can be made to correct the resonance problem.

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Appendix A
Glossary of Terms

Accelerance — see **inertance**.

Acceleration — the rate of velocity change of a mechanical system. Usually measured in units of **g** (or sometimes **G**) in English units (1 g = 386.4 in/s^2 = 32.2 ft/s^2); the international standard unit is **m/s^2**, 1 g = 9.806 m/s^2.

Acoustics — the study of sound and its interaction with the human hearing mechanism. See also sound intensity and sound pressure.

Admittance — see **receptance**.

Amplitude — magnitude of a measured signal.

Analog — describes a signal represented by a proportional electrical voltage, current, charge, etc. By association, any device which operates directly on an analog signal is an **analog device**, for example analog amplifiers and analog filters are used to condition analog signals at the front end of a digital signal analyzer.

Analog Integration — converting an analog signal representing one parameter into an analog signal representing a different parameter by using analog electronics. For example, an analog signal representing acceleration can be analog integrated once to get an analog signal which represents velocity, or integrated twice to get an analog signal that represents displacement.

Analog-to-Digital Converter (ADC or A/D converter) — a device for converting an incoming analog signal to a series of discrete digital values by sampling. Described by the number of bits it uses, (e.g., 8 bits or 12 bits), it is a key element in a digital analyzer. The number of bits sets a theoretical upper limit on the analyzer's dynamic range, equivalent to approximately 6 dB per bit.

Anchor — a term used to describe a reference mark used for measuring delta time or for marking sideband intervals in the CSI Model 2400 cursor functions.

Apparent mass — the frequency response function representing force per unit acceleration. Note: only in special cases is this function the inverse of the inertance.

Autocorrelation — a time domain function that compares a signal with delayed copies of itself, using all possible time delays, and shows at which time delays the signal repeats itself (periodicities). Although it has seen some applications in characterizing signals buried in noise, usually more information is found in the frequency spectrum of the autocorrelation which is the autospectrum (or power spectrum). Autocorrelation is a special case of cross correlation.

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Autoranging — the process of automatically adjusting the input gain of an analyzer to match the amplitude of a signal. Optimizes the use of the dynamic range of the analyzer and improves signal-to-noise ratio.

Autospectrum — another term for the averaged power spectrum of a signal. It has magnitude only, phase information being effectively discarded during processing.

Averaging — a process of summing a number of time waveforms, sample by sample, or a number of spectra, frequency by frequency, to obtain a better estimate of the mean properties of a signal in the presence of noise or other interfering signals. May be carried out in a number of ways, with or without weighting, including linear (power), negative linear, exponential, peak hold, and synchronous, linear vector types.

A-Weighting — a frequency spectrum shaping applied to frequency spectra in acoustics. The effect is designed to approximate the way that the human ear perceives the loudness of sound. Sound levels are reduced at low frequencies and at very high frequencies, where the ear is less sensitive. There are national and international standards for A-weighting.

Bandwidth — (a) the overall frequency range of an analyzer, e.g. 0 to 1000 Hz; (b) the frequency range over which a filter passes a signal without attenuation; (c) the effective frequency range represented by one line in an FFT spectrum.

Baseband — the frequency range from the low-frequency cutoff to the maximum analysis frequency for an FFT analysis.

Baud Rate — unit of speed for data transmission over a serial communications link.

Bodé Plot — in general, a dual plot of both the phase and magnitude information in a signal as a function of frequency. Often used in a more specialized way to describe the plots of synchronous amplitude and phase as functions of synchronous frequency for a machinery spinup or coastdown.

Coherence — a function of frequency which describes the degree of linear relationship between two signals. Used to assess cross-channel measurement quality, locate noise sources, and to check out transmission paths.

Compliance — or Dynamic Compliance: see **receptance**.

Correlation — see **autocorrelation** and **cross correlation**.

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CPM — cycles per minute. Favored by many in machine vibration analysis since the vibration caused by unbalance shows up at a frequency in CPM equal to the RPM of the shaft. Sixty cycles per minute (CPM) is equivalent to one cycle per second which equals one hertz.

Cross correlation — a time domain function that compares two signals using all possible time delays and shows at which time delays the two signals are strongly related. Although it has applications in transmission path analysis and system identification, usually more information is found in the frequency spectrum of the cross correlation, which is the cross spectrum.

Cross spectrum — the basic cross-channel measurement. Used most in calculating other functions such as the **Transfer Function**, **Frequency Response Function** and **Coherence**. Its magnitude measures how strongly two signals are related, frequency by frequency; its phase measures the phase shift between two signals, frequency by frequency. Whether the cross spectrum is from A to B or from B to A depends on whether the phase of channel B is referenced to channel A or the phase of channel A is referenced to channel B. Both forms have the same magnitude, but their phases are equal and opposite. Sometimes called the cross-power spectrum.

Cursor — a manually controlled marker that can be moved across a plot to indicate the amplitude at each time or frequency. See also harmonic and sideband marker.

Decibels (dB) — a logarithmic system of nondimensional units that measures the size of a quantity relative to a reference level. Any quantity can be measured in this way, as can any two quantities with the same dimensions be compared using decibel measure. Given a reference power (amplitude squared) level W_{ref}, any other power quantity W, having the same dimensions, may be expressed in decibels using the formula:

$$dB = 10 \log_{10} (W / W_{ref})$$

If a quantity X is in RMS amplitude units, and X_{ref} is a suitable reference level, the formula may be rewritten using $W = X^2$, to give:

$$dB = 20 \log_{10} (X / Xref)$$

Digital — describes a signal whose level is represented by a discrete series of numbers, in a format suitable for processing by a digital computer. The representation may be in the form of a waveform or a spectrum. By association, any device that operates on a digital signal is a ***digital device***, for example digital filters are used to filter digital signals after conversion from analog to digital form in a digital signal analyzer.

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Digital Integration — converting a digital signal representing one parameter into an analog signal representing a different parameter by using digital processing. A (digital) frequency spectrum representing acceleration can be digitally integrated once to get a (digital) frequency spectrum which represents velocity, or integrated twice to get an analog signal which represents displacement. A single digital integration of an FFT spectrum (in RMS, 0-pk or pk-pk amplitude format) is carried out by dividing the magnitude of each spectrum line by the frequency (in radians per second) of the line.

Displacement — the distance that an object moves, especially when vibrating. There are two types of vibrational displacement in common use:

(a) relative displacement: e.g., the movement of a shaft relative to a bearing surface, measured by proximity or eddy current probes.

(b) absolute displacement: as measured from a doubly integrated acceleration signal, picked up by an accelerometer on the casing of a machine. Units are mils (thousandths of an inch) and microns (millionths of a meter), which may be shown in RMS, 0-Pk or Pk-Pk formats.

Downloadable firmware (software) — refers to firmware (software) for controlling an analyzer that may be transferred to the analyzer from a computer over a computer interface. The Model 2400 has downloadable basic operating firmware, as well as downloadable applications software, including the standard FFT program supplied with every unit.

Dynamic Flexibility — see **receptance**.

Dynamic Range — the ratio between the largest and the smallest signals that an analyzer can detect when measured at one and the same time. Not to be confused with the input range of an analyzer, which depends on the available gain settings in the analyzer, nor the equivalent range of the A/D converter, which sets a theoretical upper limit. Actual dynamic range depends upon the quality of the analog electronics in the input signal conditioning stages, the number of bits and error level in the A/D converter, the jitter in the sampling clock, effects of quantization, and the accuracy of the computation.

Dynamic Stiffness — the frequency response function representing force per unit displacement. Sometimes known as effective stiffness. Note: only in special cases is this function the inverse of the receptance or compliance.

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Exponential — (a) a type of exponentially decaying window applied to transient waveform data to improve its decay rate and minimize leakage in the spectrum; (b) a type of frequency weighting used to give a better measure of the averaged value of a time-varying signal.

External sampling — using an external sampling clock signal to override the normal internal sampling clock of an analyzer. Usually uses a synchronous signal giving a fixed number of pulses per revolution of a shaft. In this way all FFT frequency components synchronous with the shaft rotation occur exactly at the center of frequency lines and may be measured without leakage using a uniform window. Nonsynchronous frequency components will appear to shift in frequency and are likely to be subject to leakage. Absolute frequency information in the spectrum is lost: each line represents frequencies which are some fixed fraction of the rotation frequency of the machine.

FFT — Fast Fourier Transform. An efficient method of computing a frequency spectrum from a sampled signal waveform, especially suitable for digital computers.

Filter — an analog or digital device which removes or attenuates unwanted frequencies in a signal.

Firmware — the software which controls or instructs the basic operating functions of CSI Machinery analyzers. So-called because this type of software is often burned in to the microchips.

Flat Top — a type of time window designed to minimize amplitude errors in the frequency range, at the expense of frequency discrimination. Mostly used for analyzer calibration at a given frequency.

Force/exponential — a window combination used to improve the quality of the analysis for an impact test (or other test with impulsive excitation). The exponential window is applied to the response channel data, to minimize leakage, and the force window to the exciting impulse channel data, to improve the signal-to-noise ratio. A force window is a short rectangular (uniform) window which brackets the impulse and forces all other data points to zero.

Frequency — number of times an event repeats itself per unit of time. Units are hertz (Hz = cycles per second) or cycles per minute (CPM).

Frequency response function — a spectrum representing the input/output relationship for a system, e.g., the vibratory response of a structure to an exciting force. The frequency response function is computed as a special case of the transfer function.

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Fundamental frequency — the first frequency in a series of harmonic frequencies. For example, the orders of shaft turning speed occur at harmonics (integer multiples) of shaft turning speed.

G (or g) — a unit of acceleration, commonly used with the English system of units; 1 g represents the acceleration due to gravity at sea level and is approximately equal to 386.4 in/s^2, or 32.2 ft/s^2 (9.806 m/s^2).

Group velocity — the rate of change of phase with frequency for the cross spectrum of two signals. If the two signals represent two measurement locations, the group velocity can be used to estimate the time of flight of a signal passing between the two points.

Hanning window — a shaping function applied to a time record before the FFT is calculated in order to smooth out end effects and reduce leakage in the spectrum. Usually the default window type to use for analyzing continuous signals because of the compromise between frequency discrimination and leakage suppression. See also **Windowing**.

Harmonic — an integer multiple of a **fundamental frequency**.

Harmonic Marker — a marker used to indicate the multiple harmonics in a frequency spectrum.

Hertz — a unit of frequency equal to cycles per second (CPS), usually abbreviated as **Hz**. Favored by many in general signal analysis because it is the international (SI) standard frequency unit. One hertz is equivalent to one cycle per second which equals 60 cycles per minute (CPM).

Impact Test — a type of test used to investigate the properties of a structure, in which the structure is caused to vibrate by an impulsive load from an instrumented hammer and the vibratory response is picked up by a vibration transducer.

Impedance — (a) ***mechanical impedance*** is the frequency response function representing force per unit velocity. Note: only in special cases is this function the inverse of the mobility; (b) ***acoustic impedance*** is a frequency response function relating the sound pressure produced by a sound source (such as a vibrating surface) per unit volume velocity; (c) ***specific acoustic impedance*** is a frequency response function representing the sound pressure per unit area of vibrating surface; (d) ***electrical impedance*** is the frequency response of an alternating current electrical system, representing the ratio between voltage and current.

Inertance — the frequency response function representing acceleration per unit force. Also known as accelerance.

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Instantaneous spectrum — the basic spectrum computed by the FFT process from a single data record. A complex-valued function, it contains information about both magnitude and phase with respect to the start of the time record.

Intensity — a quantity that measures the rate of power flow through a surface per unit area, in a given direction. Usually refers to acoustic or sound intensity, but may also be defined for vibratory power flow. May be measured with FFT analyzers by using two transducers and the cross spectrum.

Linear — describes a system with special properties that has an output spectrum directly proportional to its input spectrum and that does not generate new frequencies. The constant of proportionality is fixed, but may differ from frequency to frequency. Many systems can be analyzed as if they were linear, at least over restricted parameter ranges.

Linear averaging — a type of averaging in which the mean square magnitude of the instantaneous spectra of a number of time records is computed for each frequency. Also known as power averaging, the averaged spectrum is often expressed in the RMS format.

Linear vector averaging — a type of frequency averaging in which amplitude and phase information in each instantaneous spectrum are used to define a vector and averaged in a vector sense with spectra from other time records. Only meaningful if analysis is triggered, in which case the spectrum is identical to the FFT of the time averaged waveform.

Linear vector spectrum — a spectrum that has both magnitude and phase information, or, equivalently, a complex spectrum. Examples include an instantaneous spectrum, the cross spectrum, and a linear vector averaged spectrum.

Live-Time — a term growing in popularity, generally means showing the waveform and frequency spectrum at the same time as data is being acquired. A live-time display gives a visual impression of how a signal varies with time. Compare with **real-time**.

Mil(s) — a unit of displacement equal to one thousandth of an inch.

Mobility — the frequency response function representing velocity per unit force.

Modal Analysis — the process of modeling the modes of structural vibration, including resonance frequencies and damping, by mechanical testing, frequency analysis and computer processing.

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Negative linear averaging — a type of averaging that starts from an initial linear averaged spectrum and proceeds to compute a new averaged spectrum by subtracting the contribution of each new data record, instead of adding as in linear averaging. Very useful for subtracting background effects from averaged data.

Nonlinear — describes a system whose output is not proportional to its input.

Nyquist Plot — in general, a polar plot of the real and imaginary parts or magnitude and phase, of a complex spectrum, such as an instantaneous spectrum, a linear vector averaged spectrum or a cross spectrum. The term is also used in a special sense in the analysis of machine spinup or coastdown data to mean a plot of the synchronous magnitude and phase of one of the orders of shaft running speed in a polar format as the synchronous first order frequency changes. At any speed, the magnitude and phase in the Nyquist plot are exactly equal to the magnitude and phase in the corresponding Bodé plot.

Octave band — describes a type of bandpass filter that has a bandwidth equal to 70.7% of its center frequency. Conventionally used for analyzing sound levels, there are standardized center frequencies and filter characteristics for such filters. Adjacent octave band filters have center frequencies spaced approximately a factor two (one octave) apart, hence the name. A frequency analysis in terms of octave bands is called an octave (also whole octave or 1/1 octave) analysis. An FFT analyzer can be used to synthesize octave band filters from an FFT spectrum and thereby carry out an octave band analysis, although this is strictly only valid for steady signals.

One-third octave — like octave band, but for a bandwidth of approximately 23.1%. There are three one-third (1/3) octave bands for each octave band. A one-third octave spectrum can be synthesized from one or more high resolution FFT spectra.

Operating Deflection Shape (ODS) — the way that a machine or structure is deforming, or moving, at one or more frequencies under the action of normal operating loads. Use of multichannel analysis greatly simplifies and improves the range and quality of ODS analysis; dedicated ODS or modal analysis software can be used to generate visualizations of the operating deflection shapes.

Order — a multiple of a shaft turning frequency. The first order is the shaft frequency itself, in CPM numerically equal to the machine RPM.

Order Analysis — a frequency analysis in which the frequencies are expressed as orders of shaft frequency, either by normalizing by the shaft frequency, or by using external synchronous sampling.

Order Tracking — a measurement of a signal from a machine whose speed is changing with time, showing the level of one or more orders as a function of machine speed or time.

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Overlapped processing — a way of acquiring and processing data when using a Hanning window, in which each successive time record uses part of the previous record. This gives an increase in the rate at which the display can be updated, giving an apparent increase in live-time rate for lower frequency ranges. There is also an improvement in the smoothness of the data and the statistical error is reduced for a given data acquisition time. However, the method is inappropriate when triggering data acquisition and it causes an actual reduction in the real-time rate. An overlap of at least 50% (1/2) is required to gain benefits from smoothing the data, but the smoothest data is achieved for overlaps of 2/3, 3/4, 4/5, 5/6, etc.

Peak — (a) the overall maximum level of a signal in a given period of time. For sinusoidal (single frequency) signals, the peak level is 1.414 ($\sqrt{2}$) times the RMS level; for non-sinusoidal (multiple frequency) signals, this is no longer true, and the peak and RMS are not simply unrelated. Abbreviated as pk, 0-p or 0-pk. (b) An isolated maximum in a frequency spectrum, either due to a single-frequency component or the resonance of a system.

Peak-to-Peak — (pk-pk, p-p) the difference between the maximum and the minimum levels (positive or negative) in a signal over a given period of time. For a sinusoidal (single frequency) signal, the peak-to-peak level is always 2 times the peak level and 2.828 ($2\sqrt{2}$) times the RMS level. For non-sinusoidal (multiple frequency) signals this is no longer true and there is no simple relationship between peak-peak, peak and RMS levels.

Peak hold — a type of averaging in FFT analysis in which the maximum level is retained for each frequency line over all data records processed.

Period — time required for one complete cycle of a periodic signal.

Periodic signal — an ideal signal that repeats itself exactly after a fixed finite interval of time and exists for all time. Although not possible in the real world signal, many signals behave like periodic signals for a certain length of time, for practical purposes. Fundamental concept behind FFT analysis.

Phase — (a) the relationship between the angular location of the high spot and heavy spot for an unbalanced rotor; (b) the angular relationship between the peak in the vibration at a synchronous frequency and a phase reference (tachometer) pulse, for a rotating machine; (c) the delay between two signals at a given frequency, expressed as a fraction of a cycle, usually in degrees.

Power averaging — see **linear averaging**.

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Power spectrum — a spectrum of a signal formed as the mean square average level of a number of instantaneous spectra. Often displayed in RMS format, by taking the square root of the mean square level at each frequency.

Power Spectral Density (PSD) — a representation of the power in a signal that compensates for the bandwidth of the analysis. In FFT, PSD is computed from a power spectrum (in units of power = $(\text{amplitude})^2$), by dividing by the bandwidth of each line in hertz. The units of PSD are then $(\text{amplitude})^2/\text{Hz}$. Sometimes, an RMS format is used, in which case the units are $(\text{amplitude})/\sqrt{\text{Hz}}$.

Pre-/Post Trigger — triggered data acquisition using a delay so that the time record starts before (pre-trigger) or after (post trigger) the trigger event. The Model 2400 can use trigger delays from 8 time record lengths before the trigger to 100 time record lengths after the trigger.

Real-time rate — refers to the frequency at which the time an analyzer takes to compute an FFT is equal to the time required to acquire the data. Commonly used as a measure of the speed of an analyzer and equally commonly confused with the rate at which the display is updated. Overlapped processing reduces the actual real-time rate.

Receptance — the frequency response function representing displacement per unit force. Also known as admittance, (dynamic) compliance, and dynamic flexibility.

Resolution — the frequency range represented by one line of an FFT spectrum. Found by dividing the maximum analysis frequency by the number of lines. The resolution in Hz is equal to the inverse of the data record length in seconds.

RMS — root mean square; as applied to a dynamic signal such as vibration or sound refers to an averaged level of a function obtained by averaging the square of the signal level over a period of time (or number of data records), then taking the square root of the result.

RS232 — a serial, asynchronous communications standard; a type designation for cables used to connect communications ports on a computer with other digital devices such as digital analyzers, printers and modems.

Sideband — a frequency component that represents the effect of modulation on a signal. If a modulated signal has more than one component, each component will show sidebands. A sideband is spaced off from the frequency of the modulated signal by an amount equal to the modulating frequency. If the modulating signal has multiple components or if there is frequency modulation, the sideband pattern may be very complicated including sum and difference frequencies between the sideband component frequencies (intermodulation effects).

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Sideband Marker — a marker used to indicate the sidebands around a center frequency marked by setting a mark, then highlighting an adjacent frequency component.

Signal-to-noise ratio — the ratio of the power of the signal to the power of the background noise effects in a measurement, usually expressed in decibels. In a signal analyzer, the signal-to-noise ratio is typically improved by increasing resolution or the number of averages, among other factors.

Software — computer programs for calculating functions or controlling digital devices with a digital computer.

Sound — vibratory movement of the air, or some other conducting fluid, characterized by a compressive wave mechanism with a constant speed of propagation in a homogeneous unrestricted medium.

Sound Pressure — the pressure exerted by the movement of fluid particles in a sound wave.

Spectrum — the frequency domain representation of a signal. In practical measurements, the spectrum usually displayed as a plot of magnitude against frequency over a limited frequency range. See also **cross spectrum**, **power spectrum** and **linear vector spectrum**.

Synchronous averaging — a type of averaging in which successive time records are averaged together without computing a frequency spectrum. If the analysis is triggered synchronously from a rotating shaft or some other periodic event, the averaged waveform will emphasize the synchronous components of the signal and suppress the asynchronous components like noise and background effects. The spectrum of the synchronously averaged signal is a linear vector averaged spectrum of the data, having both magnitude and phase information.

Tachometer — device that generates a pulse signal corresponding to the revolution of a shaft, used to measure turning speed. A single pulse per revolution may be used to trigger data acquisition synchronously with shaft rotation.

Time record length — the time required in FFT analysis to acquire the number of samples required to obtain a given number of lines at the sample rate required to achieve the maximum analysis frequency selected.

Transfer function — a spectrum representing the relative magnitude and phase of two signals. For two signals A and B, the transfer function from A to B is the ratio of the cross spectrum from signal A to signal B, divided by the autospectrum (power spectrum) of signal A. Its phase is equal to the phase of the cross spectrum.

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Transient — a time-varying signal of finite duration, i.e. having a definite start and finish. May refer to an impulsive signal, such as a hammer blow or the vibration signal from a machine coastdown or spinup. Such signals have finite energy, unlike periodic signals.

Trigger — the control signal for starting and stopping data acquisition. May be based on an incoming measured signal, an external pulse signal, or an internal clock.

Uniform window — a type of window used for analyzing a signal without shaping. Subject to leakage and amplitude errors if the frequency components are not centered on a line in the spectrum. Suitable for transient signals wholly contained within the analysis time record length and when using external sampling. Also known as a Bartlett, Boxcar, or Rectangular window. Compare Hanning window.

Velocity — the rate of change of displacement of a mechanical system. Units are inches per second (in/s or ips) in English units and m/s, cm/s or mm/s in SI units. Can be measured directly with a velocity pickup or by integrating an acceleration signal from an accelerometer.

Vibration — the oscillatory motion of a mechanical system about a mean position.

Waveform — analog or digital representation of a signal displayed as a plot of level against time.

Windowing — a process of applying a weighting to a waveform signal before computing the FFT in order to minimize leakage and/or the picket fence effect that gives misleading spectrum levels. See also **Hanning, Uniform, Exponential, Force/exponential.**

Zoom — a frequency analysis at higher resolution than the baseband spectrum over a limited frequency span in order to see more detail. There are two types: nondestructive zoom and real-time or true zoom. The latter involves re-analyzing the signal (destructive), using frequency translation and digital filtering to obtain the results. Nondestructive zoom involves acquiring more samples in the first instance, giving a potentially higher resolution anywhere in the baseband frequency range. More detail can then be seen merely by expanding the frequency scale in the region of interest.

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